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# POWER

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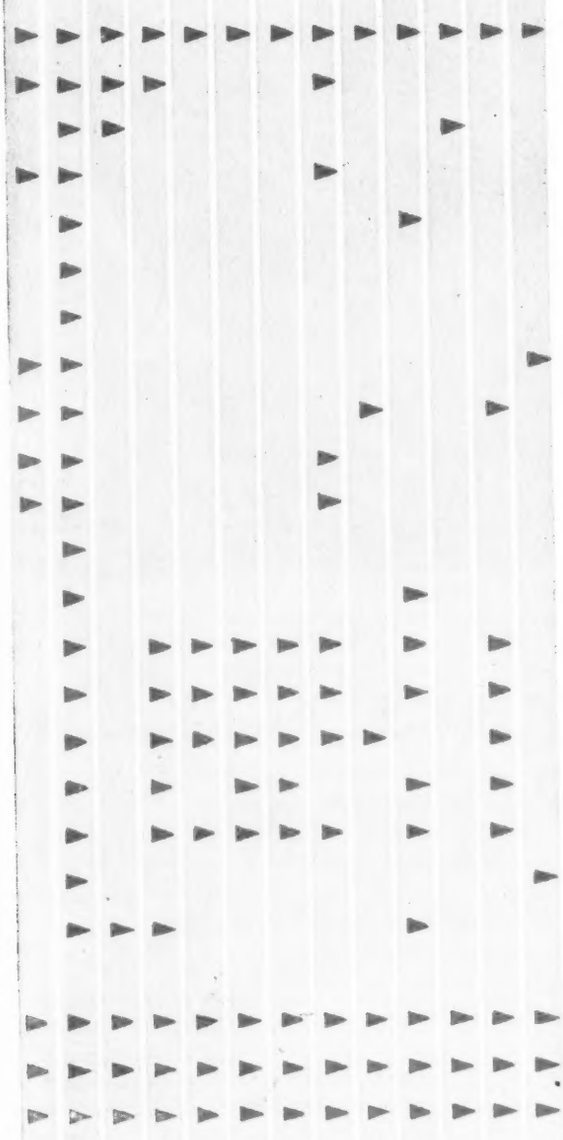

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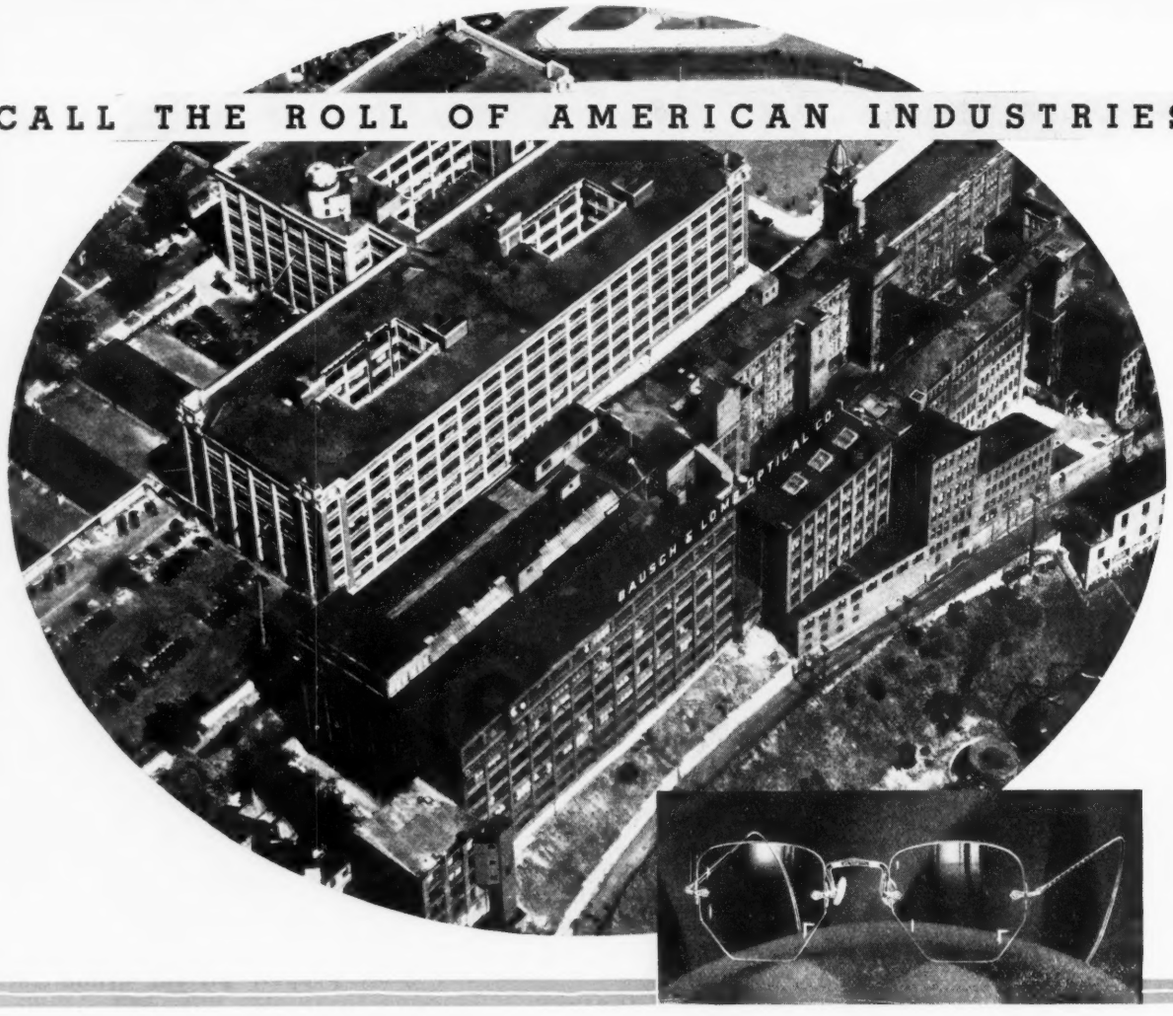


# ANALYZING CONDITIONS MAKES IT EASY TO "SEE" THE PLACES FOR

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**POWER**

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PHILIP W. SWAIN, *Editor*

# HORSEPOWER ON PARADE

MAN'S first prime mover was his own $\frac{1}{10}$-hp. muscular motor. Then he "got a horse" and glorified in this 900% increase in strength. Yet, the horse, companionable worker and swift conveyance in war and peace, could not keep pace. So man learned long ago to harness the winds to propel his ships. Next he put cloth sails on a wheel to make a windmill and wooden sails on another to build a waterwheel.

For centuries these two—muscles and sails—were the only conceivable prime movers. Whatever work was done, they did. And because their power was small, man's works were always puny except where tens of thousands of sweating slaves were regimented to build great pyramids, aqueducts and walls. Three sensational inventions were needed before modern power could be born—steam as a motive force, the piston, the crank.

They may look simple in retrospect, but these inventions, astounding departures from all previous experience, must rank among the very greatest achievements of the human mind. With the existing wheel and sail, they led inevitably to the steam engine, the steam turbine, the piston pump, the centrifugal pump and the rotary fan. Add the discovery that air, heated by internal combustion, can expand and work like steam on a piston, and you have all the elements of today's spectacular horsepower parade.

To these basic discoveries generations of skilled mechanics and persistent engineers have added details and improvements in infinite variety. Where do we stand after all these outpourings of inventive genius? We stand in a power age where human horsepower is comparatively worthless except to push buttons, throw levers and guide fountain pens. Yet man's load has not disappeared; it has merely shifted from brawny shoulders to the part of the body that lies above the eyes.

A whole new breed of men has been trained merely to direct these prime movers. Simply to be a good judge of these new varieties of "horseflesh" demands a lifetime of study and experience. This, then, is the new "white man's burden"—knowing how to choose the right prime mover for a given time and place.

Today's swift progress of design makes the job doubly difficult. And the same swift pace doubles the reward of the engineer who can select intelligently. For change produces obsolescence, and obsolescence is but another spelling of opportunity.

Clearly, the times call for a broad survey of the existing prime-mover situation—a summary of types and subtypes—an unbiased listing of their advantages and disadvantages for various applications—and, above all, a clear presentation of the philosophy of selection.

So here are forty pages devoted to this single job. Special sections, distinguished by color, deal separately with water turbine, steam engine and turbine, diesel and gas engine. Additional articles cover such general problems as governing, speeds and pressures, "tops." "Which Prime Mover?" brings all together in a single picture.

Here are the ABC's, and some of the XYZ's, of the units that lie at the very heart of the power engineer's job. Tying together and reinforcing the odds and ends of his own prime-mover experience, they point the way to cheaper power through wise investment.

# WHICH PRIME MOVER?

IF IT were possible for this article to tell you just what kind of prime mover to select for a given set of conditions, the practical thing to do would be to give you the answer and have it over with. One may be able to pick an automobile that way, but it won't work for a power plant. Which is one of the reasons why the selection of prime movers should be made by engineers rather than by plumbers or statesmen.

However, this article can help you on this job: *first*, by roughly bracketing the fields of the various prime movers; *second*, by indicating the "natural" applications for each general type; *third*, by setting down certain fundamental principles that apply in all cases, *fourth* by showing how to organize the job of applying these principles.

The emphasis in this presentation will be chiefly on the choice *among the primary types*, for example, between steam turbines and diesels. To get more light on *which kind* of diesel or *which kind* of turbine of steam engine, turn to the section discussing that particular prime mover.

### Habits with Reasons

If we look around we find each type of prime mover practically excluded from certain applications. There may be exceptions but, in general, large central stations never install steam engines and rarely install diesels. (This discussion is confined to American practice). It is increasingly rare to install any steam prime mover for electrical generation where all exhaust must be wasted to atmosphere, unless the price of purchased current is exorbitant.

It is questionable practice to install diesel engines in process industrials with large power and process loads so timed that the power delivered by the diesel could have been generated largely as a byproduct of process and heating steam.

Approaching the problem from the other side, we can list certain fields for which each prime mover is particularly fitted and universally or widely used. Excepting the mercury turbine, the steam turbine is the only prime mover ordinarily considered for the large central station and certainly the only one for the large *steam* central station. For oil-line pumping, and natural-gas-line compressing, gas and diesel engines have (and should hold) a practical monopoly, except where motor drive is economical.

These are some of the exclusive fields of application. Next come the overlapping fields. For the small central station, the primary choice is between steam turbine and diesel, with the diesel probably gaining ground at the present time. This refers to plants under 5,000 kw. principally.

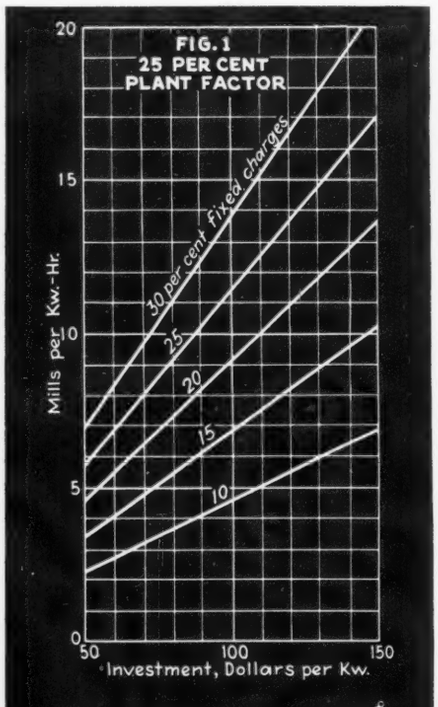
Industrial plants, as a whole, offer a good field for every type of prime mover, but with the steam turbine gaining an advantage over the engine as plant size increases. Other things being equal, the availability of cheap condensing water favors the turbine over the steam engine in cases where the load is substantial and where it is not possible to generate all power requirements as a byproduct of heating and process steam.

If the intention is to generate all power as a byproduct, the choice between engine and turbine depends on several factors. First comes size of plant. The small non-condensing engine (500 hp. or less), particularly the uniflow, is considerably more efficient than the small non-condensing steam turbine. This is important if the ratio of process steam load to power load is small, or if the loads do not coincide; otherwise not. On the other hand, if conditions favor condensing-bleeding operation the turbine is a natural choice.

The non-condensing turbine is generally somewhat cheaper than the non-condensing engine and takes less floor space. On the other hand, the advantages of the engine for service in buildings and institutions, exhausting to extensive building and waterheating systems, are indicated by its wide use for such service.

In industrial plants requiring several process pressures, the steam turbine with multiple bleeding avoids complicated systems of superimposed engines.

In the past, the absence of oil from turbine exhaust was urged as an advantage over the steam engine wherever the exhaust was to be mixed directly with process water or product. This is still a point to consider, but less important than formerly, because efficient oil separators are now available. High starting torque is a great



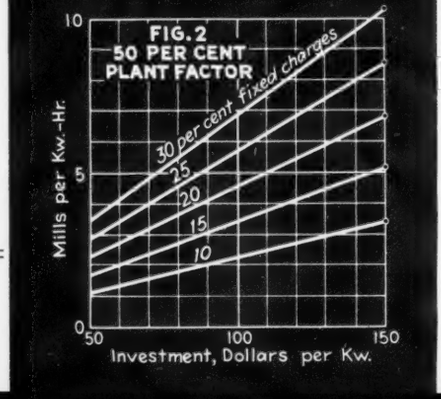




Fig. 1—Fixed charges per kw.-hr. for 25% plant factor. Fig. 2—Fixed charges per kw.-hr. for 50% plant factor

advantage of the steam engine for mechanical drive of certain loads.

The inability of most steam engines to handle high superheat is not ordinarily important in industrial plants, except for high-pressure condensing operation. Even with fairly high throttle pressures, back-pressure operation does not require very high steam temperatures.

Very few steam engines have been installed in industrial plants for really high-pressure operation—500 lb. and up. Yet this would appear to be a rich field for steam engines, particularly in plants generating from 200 to 1,000 hp. Engines for such service can be made far more efficient than steam turbines and thus make possible a greater recovery of byproduct power from a given flow of process steam. Handling the pressure presents no great difficulties, while experiments going on in Europe indicate that problems of lubrication and piston packing can be solved. It is to be hoped that the demand for units of this type will lead to their promotion in America, probably in the form of multiple-cylinder, vertical, high-speed units resembling automobile engines.

Whole books can be written about the field of application of the diesel engine. There has been a lot of smoke and some fire on both sides of this diesel argument. Undoubtedly the costs claimed in some cases have been excessively low. Yet, to an impartial observer three facts are evident:

*First,* the diesel has had many economically sound industrial applications for years. *Second,* diesel advocates show a constantly increasing willingness to seek out and publish true costs, including fair allowances for fixed charges, lubrication, maintenance, etc. *Third,* while the discussion has been going on, the engines themselves have been getting constantly better and less costly. Today the diesel engine is thoroughly established as a prime mover. It cannot be "laughed off," although it is still often misapplied.

It is of course quite easy to overemphasize the thermal-efficiency angle. Thermally the diesel is extremely efficient, surpassing the finest and largest steam central stations. Yet, heat unit for heat unit, its fuel is generally much more costly. At 4¢ per gallon, diesel oil costs about 30¢ per million B.t.u. Coal at $4 per ton costs about 15¢ per million B.t.u. Therefore the fuel cost per kw.-hr. is generally considerably less for a large, modern coal-fired central station than for a diesel plant.

Yet steam central station efficiency falls off rapidly in small sizes while investment per kilowatt go up. The small diesel plant costs little more per kw. than the large and is almost as efficient. Standby losses of the steam plant are eliminated.

Fuel cost is the easiest part of diesel cost to figure, because it varies but little with engine size. For almost any sized plant, and any plant factor within reason, the yearly average will lie between 8 and 12 kw.-hr. per gallon. Test figures, of course, are close to 12 kw.-hr. Price of diesel oil ranges from 2¢ to 6¢ per gallon, the national average being around 4¢ at present. Lubricating oil is a substantial expense with diesels, 8 to 20% of fuel cost.

At the present time a 700-kw. diesel plant can be built complete, including building but not land, for about $135 per kw. This compares favorably with the cost of a complete condensing steam plant of the same size. Further reductions in the cost of diesel plants may be expected.

Unless coal is very cheap, or oil high, it is very difficult to figure out a complete steam plant under 2,000 kw. to compete with a diesel plant if the steam plant is exclusively for power generation. But, at the other extreme, we have the moderate-sized steam plant with complete byproduct power generation, where the diesel hasn't a "look-in" because the steam prime mover's fuel consumption is only about 1 mil per kw.-hr., and little of the investment or labor, outside of that connected with the steam prime mover, is chargeable to power.

In between is the debatable ground where some large fraction of the power may be generated as a byproduct, and where both diesel and steam must be considered. In many such cases a combination of steam and diesel will give the cheapest power. In this team the steam partner skims the byproduct-power cream, while the diesel pulls the straight excess-power load. In this case the diesel also spells the steam plant on weekends, holidays, or for the summer (in building plants).

Fig. 5 will be helpful in giving a

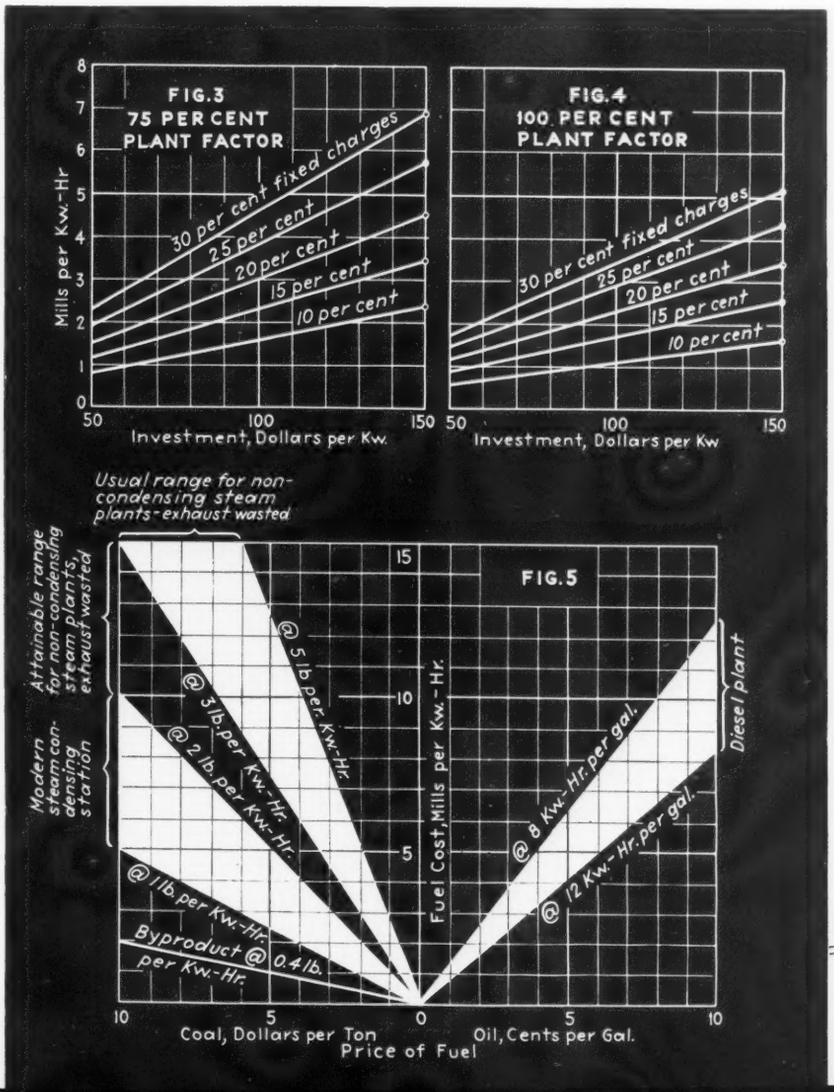



Fig. 3—Fixed charges per kw.-hr. for 75% plant factor. Fig. 4—Fixed charges per kw.-hr. for 100% plant factor. Fig. 5—Fuel cost of 1 kw.-hr. for diesel plant and coal-fired steam plant. See page 296 for explanation of the fuel rates indicated

general idea of the fuel cost of power generated in various ways. The vertical scale is fuel cost of power in mils per kw.-hr. Fuel-price scales read both ways from the center; coal to the left, oil to the right. On the left only coal-fired steam plants are considered; on the right only diesel plants. The assumed fuel rates per kilowatt-hour are not based on full-load test data, but rather represent about what an average plant put in today would do in the way of over-all yearly performance.

The coal consumption of modern central stations is carried up to 2 lb. per kw.-hr. to include stations of small size. It is believed that range from 3 to 5 lb. for steam plants exhausting to atmosphere is about what the average new plant exhausting to a heating system would figure today if all coal were charged to power (equivalent to atmospheric exhaust). Such plants could, however, be built for coal rates as low as two pounds if plants were large and throttle pressures very high.

The average industrial steam power plant makes part of its power on the lower 0.4-lb. "byproduct" line and part in the white atmospheric-exhaust zone between 3 and 5 lb. Sometimes a third part is made in a condensing zone corresponding to, say, 2 lb. per kw.-hr. The tremendous effect on fuel cost of changing the percentage of byproduct power is shown in Table I, based on the assumption that byproduct power takes 0.4 lb. of coal, condensing power 2 lb., and atmospheric-exhaust power 0.4 lb. Coal is assumed to cost $5 per ton.

Table I—Effect of Byproduct Power Generation on Fuel Cost of Power

| Percent as by-product | Remainder condensing | | Remainder atmospheric exhaust | |
|---|---|---|---|---|
| | Lb. Coal per kw.-hr. | Mils per kw.-hr. | Lb. Coal per kw.-hr. | Mils per kw.-hr. |
| 0 | 2 | 5.0 | 4 | 10.0 |
| 10 | 1.84 | 4.6 | 3.64 | 9.1 |
| 20 | 1.68 | 4.2 | 3.28 | 8.2 |
| 30 | 1.52 | 3.8 | 2.92 | 7.3 |
| 40 | 1.36 | 3.4 | 2.56 | 6.4 |
| 50 | 1.20 | 3.0 | 2.20 | 5.5 |
| 60 | 1.04 | 2.6 | 1.84 | 4.6 |
| 70 | 0.88 | 2.2 | 1.48 | 3.7 |
| 80 | 0.72 | 1.8 | 1.12 | 2.8 |
| 90 | 0.56 | 1.4 | 0.76 | 1.9 |
| 100 | 0.4 | 1.0 | 0.40 | 1.0 |

The economics of water power have been studied intensively by the utilities and a few large industrials. As a matter of general information, engineers concerned primarily with steam and diesel power should know their general conclusions, briefed as follows:

1. There are not many sites left where a new hydro plant, not interconnected with steam plants, will pay.

2. In spite of this, many recent hydro plants, particularly in the East, have proved very good investments.

3. This good result is possible where the hydro plant is tied into a system with steam plants, has storage for at least one day of minimum stream flow, installs considerably more generating capacity than can be operated continuously by the minimum stream flow.

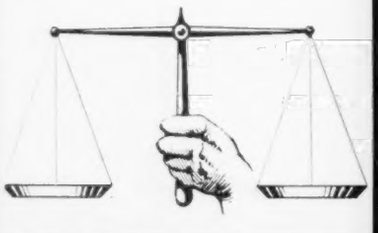
Turning back to steam and diesel, it will be granted that fixed charges are often as important as fuel costs in selection—sometimes more so.

Plant factor affects the investment that can be made quite as much as does the fixed charge rate. "Plant factor" is the ratio of the total kilowatt-hours produced in a year to what would be produced if all machines operated at 100% for the full 8,760 hr. of the year. If you cut this factor in half, the burden of fixed charges on each kilowatt-hour is doubled. Figs. 1, 2, 3 and 4 give the actual fixed charges per kw.-hr. for all investments from $50 to $150 per kw., fixed-charge rates 10 to 30% and plant factors of 25%, 50% and 75% and 100%.

To see how this works out, suppose the plant factor is 25% and the fixed-charge rate is set at 25% by company policy. Suppose also that one plant is available at $100 per kw., and another (more efficient) at $150. The difference in this extreme case amounts to 5.7 mils per kw.-hr., which is the saving in operating expense the more costly plant would have to make to justify itself.

For quick reference, the tables at the bottom of this page set down the conditions that favor one prime mover over another. To avoid misunderstanding, it should be made clear that each condition is listed merely as one of many affecting the final choice. Thus, a diesel is *more likely* to be usable in a small plant than in a large plant, but this does not imply that diesel is better than steam for small plants. Sometimes it is, sometimes not.

High fixed-charge rates are listed as favoring steam "except in little plants." This follows from the well-known fact that a large diesel plant costs more than a steam plant of the same size. If the plant is so small that diesel investment per kilowatt is less than for steam, the effect of high-fixed charged rates is, of course, reversed.



**CONDITIONS FAVORING**

| STEAM | DIESEL |
|---|---|
| Larger plant | Smaller plant |
| High fixed-charge rates (except in little plants) | Low fixed charge rates (except in little plants) |
| Cheaper coal | More costly coal |
| Cheaper boiler oil | More costly boiler oil |
| Costly diesel oil | Cheaper diesel oil |
| Condensing water available | Little condensing water |
| Much process steam | Less process steam |
| Continuous operation | Intermittent operation |
| No stack limitations | Stack limitations |
| No space limitations | Space limitations |
| | Semi-portable plant |
| | Temporary plant ("load builder") |

**CONDITIONS FAVORING**

| STEAM TURBINE | STEAM ENGINE |
|---|---|
| Larger plant | Smaller plant |
| Lower back-pressure | Higher back-pressure |
| Lower throttle pressure (small plant – see text) | Higher throttle pressure (small plant –see text) |
| Condensing water available | Cond. water not available |
| Low starting-torque load | High starting-torque load |
| Multiple process pressures | Single process pressure |
| Lack of floor space | Ample floor space |
| A.C. generation | D.C. generation |
| High-speed drive | Low-speed drive |

# WATER POWER

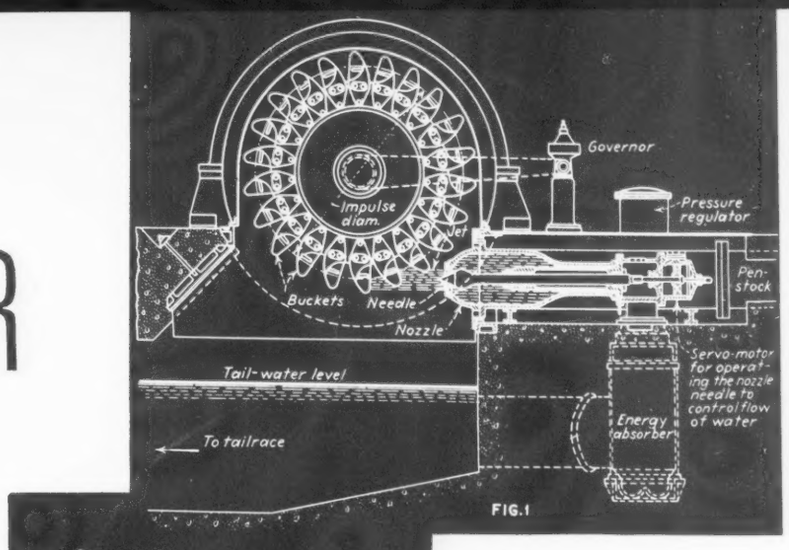



FIG. 1

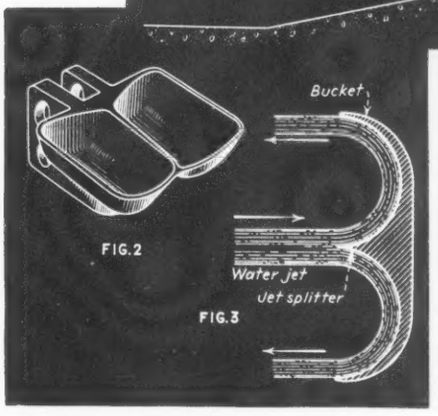



FIG. 2




FIG. 3

Fig. 1—Impulse wheels operate with a water jet striking buckets on their periphery.

Fig. 2—Pelton buckets were practically rectangular in cross-section. Fig. 3—Theoretically the jet should turn through 180 deg. in the bucket for best efficiency

MORE than twenty centuries ago, man's needs for power exceeded his own strength and that of any animal he could tame. So the Chinese and Egyptians, among others, began to tap the resources of nature—they built crude waterwheels with which water pumped itself for irrigation. You can find similar wheels today, doing the same job, but you can also find the products of two thousand years of development, wheels at Boulder Dam that develop the power of 115,000 horses, wheels at Dixence, Switzerland, that do the work of 50,000 horses by taking the energy out of a column of water more than a mile high! (See Table.)

Dozens of types of wheels have been developed and redeveloped, but power wheels today fall in three general classes: Pelton, Francis and propeller. The first handles high heads by causing a jet of water to strike buckets on the periphery of the wheel and is called an impulse type, Fig. 1. The other two handle medium and low heads. These have curved vanes on which the water acts around their entire periphery and are called reaction types.

Water can develop power by its weight, velocity, pressure, or a combination of velocity and pressure. Water's weight is still used in the overshot, where it flows from a high flume into buckets on the wheel rim. Modern wheels use velocity, or velocity and pressure.

Impulse wheels, using velocity alone, first had flat wood or iron plates around the rim for the water to strike, but the water splashed around and didn't develop much power. So Pelton tried hemispherical cups with the jet striking them in the center. Efficiency promptly increased. But one of his jets accidentally hit one side of the bucket and discharged at the other. Efficiency jumped some more. So he built new buckets like half cylinders side by side, with the jet splitting at their joining edge, going half to each bucket, Fig. 2. Theoretically for highest efficiency the jet should turn 180 deg., Fig. 3, but then it hits the following bucket and reduces efficiency. So bucket shape had to be changed to something like Fig. 4. Doble took Pelton's half cylinders and made them spoon-shaped, or ellipsoidal, Fig. 5, and increased efficiency again.

Better grades of buckets are now made of cast steel, bronze, or stainless steel, which are ground on their inner surfaces to make the "splitter" a knife edge and give the water a smooth path. This is necessary for high efficiency, to prevent eddy currents in the buckets and pitting of their surfaces. As shown in Fig. 1, the flow of water to the buckets is controlled by a needle in the nozzle that is under the control of the governor.

Usually, one jet impinges on an impulse runner on a horizontal shaft, thus only part of the buckets are working at any given time and wheel diameter must be much larger for a given power output than if all buckets were used constantly. This means that it must be run at a comparatively low speed. Two or more jets permit a smaller wheel, increasing speed and multiplying power output by the number of jets. But they complicate the nozzle arrangement and control and usually reduce efficiency because of interference, unless the water is properly guided away from the wheel. With more than two jets, it is practically necessary to use a vertical-shaft runner. Six units on Salt River, Ariz., have six nozzles each, develop 1,000 hp. under 111-ft. head at 94 r.p.m. Normally a reaction wheel would be used for such a moderate head, but this water carries so much silt that the engineers felt an impulse type would wear less.

Now to this matter of wheel speed and bucket shape. Peripheral speed of the wheel at its impulse diameter, Fig. 1, is fixed theoretically at half the velocity of the jet. Actually, its speed is limited to about 0.47 times jet velocity. Therefore, the larger it is in diameter, the lower its speed must be in r.p.m. For example, a wheel with a 60-in. impulse-circle diameter under 1,000-ft. head at the nozzle will run at 450 r.p.m. If impulse-circle diameter is halved, speed is doubled. Power that the wheel is to develop, and other design factors, limit its diameter.

For best efficiency, jet cross-sectional area should not exceed 0.1 times the

projected area of the bucket, and bucket width should not be less than three times jet diameter, but can be almost any width above this. For practical reasons, wheel diameter is fixed at not less than 12 times jet diameter. These all limit minimum wheel diameter, and therefore the maximum speed for a given output.

Europe has used multiple-jet impulse wheels extensively, but we have stuck pretty closely to single-jet designs, although in larger capacities we use double runners with the generator between them. These retain the simplicity of the single-runner unit, but have only two bearings (runners are overhung) and can be run at a considerably higher speed than an equivalent single-runner unit, with consequent reduction in cost. They also reduce the power in a single jet, an advantage.

Impulse wheels, except in smaller capacities, are generally used for heads greater than 1,000 ft. (See Table.) This is particularly true today, for Francis wheels can now operate under heads of 700 to 1,100 ft.

## Reaction Wheels

Reaction wheels, in two general classes, Francis, Fig. 6, and propeller, Fig. 14, are used for heads from about 5 ft. to over 1,000 ft. The propeller type, first used in 1916, has since practically taken over the bracket below 60-ft. head. Maximum head for this type of wheel is 106 ft. thus far, at River Shannon, Ireland.

James C. Francis, a New Englander, developed in 1849 an inward-flow wheel that developed 80% efficiency. Many similar types were developed before him, and have been since, but his name has been given the type. These wheels have curved buckets held between a crown plate and a runner band, Fig. 6. Bucket shape varies with head, Fig. 9 showing one for fairly high head (800-900 ft.), Fig. 10 for around 100 ft., and Fig. 11 for low heads of 10 to 50 ft. As head decreases, the buckets are given more flare outward from the crown plate. This carried to the ultimate and omitting the runner band approaches the propeller wheel, Fig. 14. Wheels have been built that look something like Fig. 11 with the band removed.

A complete Francis turbine, Fig. 8, has wicket gates that lap over each other as in Fig. 12, around the runner and supported in bearings in a cover plate and a base ring. These are opened and closed by the governor to control flow of water to the turbine.

For heads up to 20 ft., the turbine may be set in an open flume. Modern installations, particularly larger ones, usually have a spiral casing, Fig. 7, around the turbine. For heads up to about 60 ft. the spiral casing is generally formed in the powerhouse concrete foundation, for heads up to 350 ft. plate steel and above 350 ft.

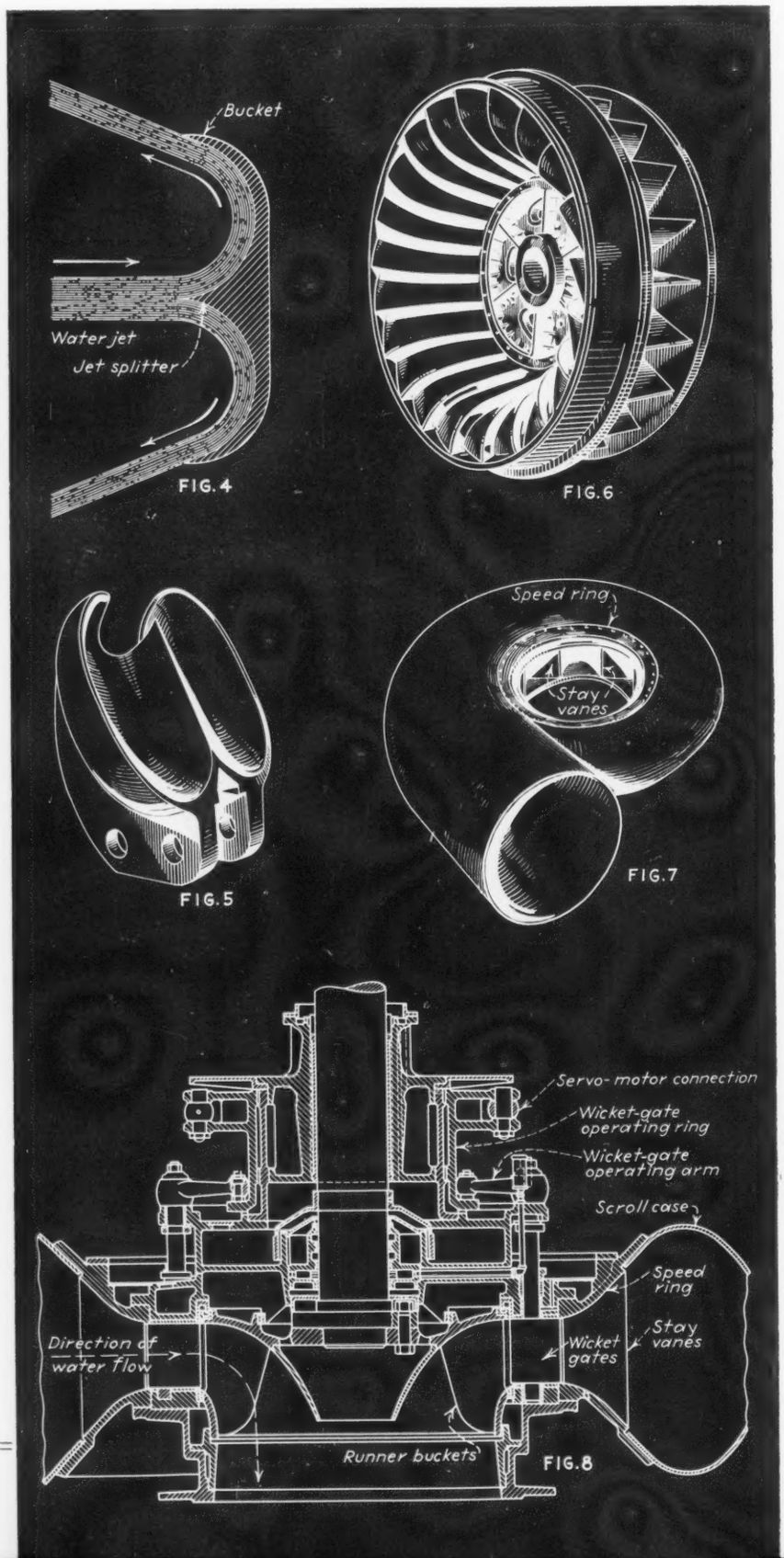



Fig. 4—To prevent its striking the following buckets back the jet is turned slightly less than 180 deg. Fig. 5—Pelton-Doble bucket for impulse wheel. Fig. 6—A large Francis-type runner. Fig. 7—Scroll case for Francis wheel. Fig. 8—Cross-section through a Francis turbine assembly

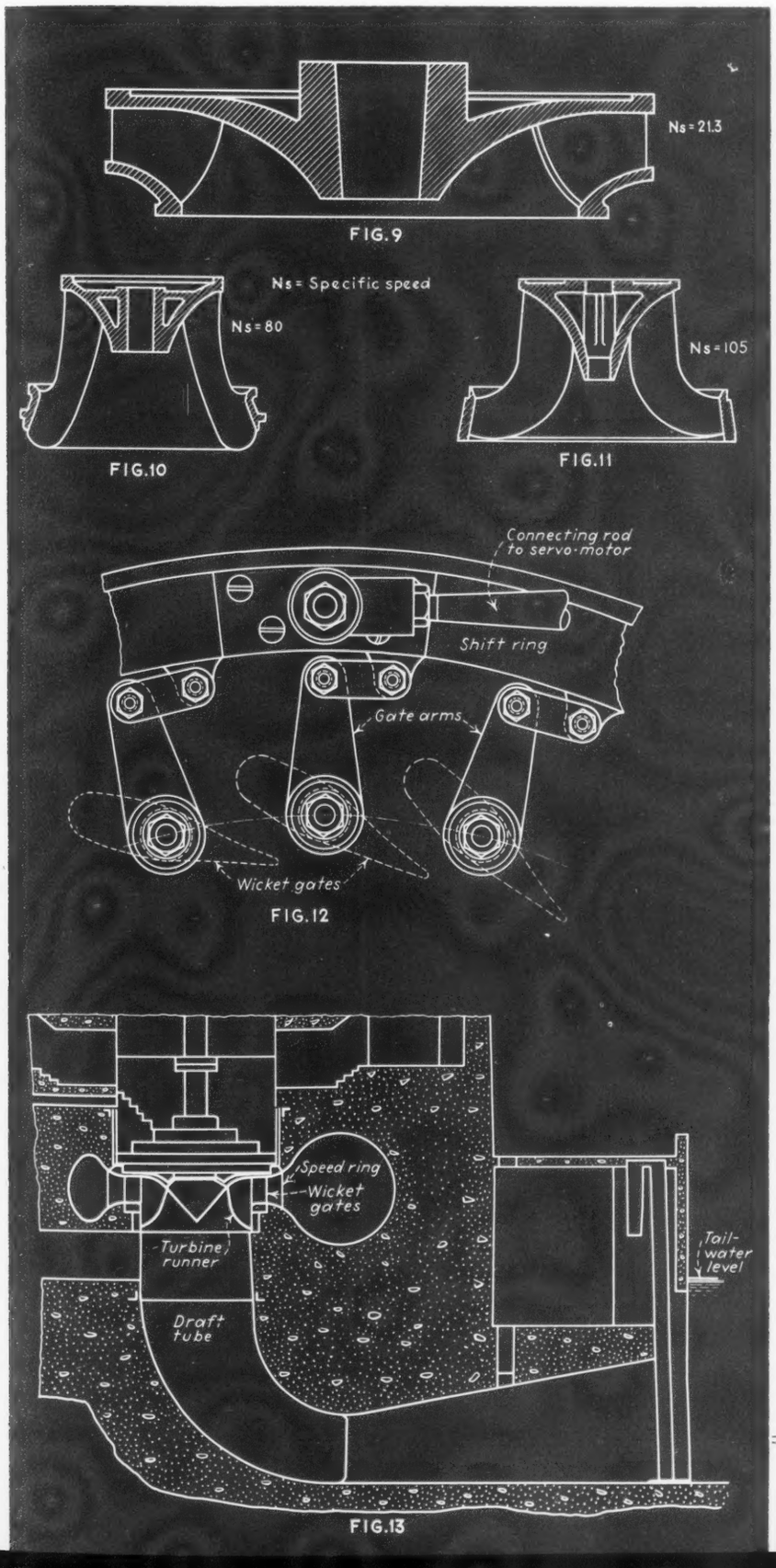
cast steel is used. Between scroll case and wheel gates is usually a speed ring with stay vanes, the first anchoring the turbine to the scroll case, the vanes smoothing out water flow and acting as columns to take pressure stresses. In vertical installations, stay vanes also help to transmit generator weight to the foundation.

Water leaving a runner can produce considerable power because of its velocity and the height of the turbine above tailwater. Part of this power is regained by a draft tube (usually the elbow type of Fig. 13) below the turbine to reduce pressure below atmospheric.

Either vertical or horizontal-shaft turbines may be used, although most modern hydro-electric units are vertical-shaft arrangements because of their compact design and development of thrust bearings to carry the load.

Reaction turbines should run as fast as consistent with other considerations, for the higher the speed, the lower the cost of unit. As in Pelton wheels, the smaller the wheel diameter for a given head, the higher its speed. Before introduction of propeller runners, it was common in low-head installations to use two or more Francis runners on the same shaft—six to eight on a horizontal shaft and up to three on a vertical one.

The propeller wheel, Fig. 14, made possible the use of specific speeds up to over 200. This wheel is also called an axial-flow type. Water flows horizontally into the turbine above the runner and turning flows axially through the blades, Fig. 15. The Francis type, Fig. 8, is sometimes called a mixed-flow turbine because the water flows radially into the wheel and then turns and flows axially.




## Propeller Wheels

Propeller wheels may have fixed or adjustable blades. Efficiency of a high-specific speed, fixed-blade runner falls off rapidly on either side of a maximum. To avoid this, propeller wheels may be built to change blade angle to maintain efficiency over a wide range of load and head. The automatically adjusted blade design is called a Kaplan type and has its blade angle controlled by the governor. One recent design utilizes the change in volume of water supplied to the runner to adjust blade angle.

Fig. 16 shows efficiencies obtained with a propeller runner with constant head and blades adjusted for 16.5- (Curve *A*), 22- (Curve *B*) and 28-deg. (Curve *C*) pitch angle. These angles give peak efficiency at 65, 80 and 95% load, respectively. If runner-blade pitch is adjusted automatically as load changes, Curve *D* results, fairly flat over a wide range.

Each blade of a Kaplan turbine (Fig. 15) is supported in the hub by two bearings *B*, and connected by a short crank arm and link to a cross-

Figs. 9, 10 and 11—Cross-section through a high-head, a medium-head and a low-head Francis runner, respectively. Fig. 12—Section of the wicket-gate mechanism for controlling water to reaction turbines. Fig. 13—Elbow draft tube used on reaction-type turbine

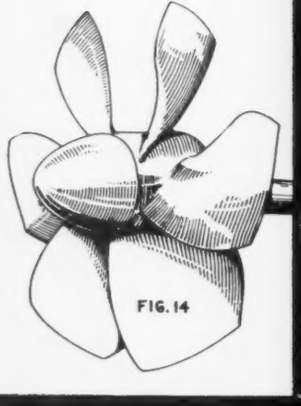
head *C*. A heavy rod running through the center of the turbine shaft connects crosshead and servo-motor piston, The inside pipe of a pair running through the generator shaft connects to the underside of the servo-motor piston by a crown plate and ports through the piston. The upper end of this pipe is plugged and attached to the compensating rod on the control valve. The outer pipe connects to the top end of the servo-motor cylinder. Pressure oil to the servo-motor is controlled by a 4-way valve in turn controlled by the governor. As the governor adjusts the turbine gates for a change in load, it also operates the 4-way valve, thus causes the servo-motor to change blade angle. The governor opens the 4-way valve, but it is closed by movement of the servo-motor piston connected to the inner pipe and compensating rod.

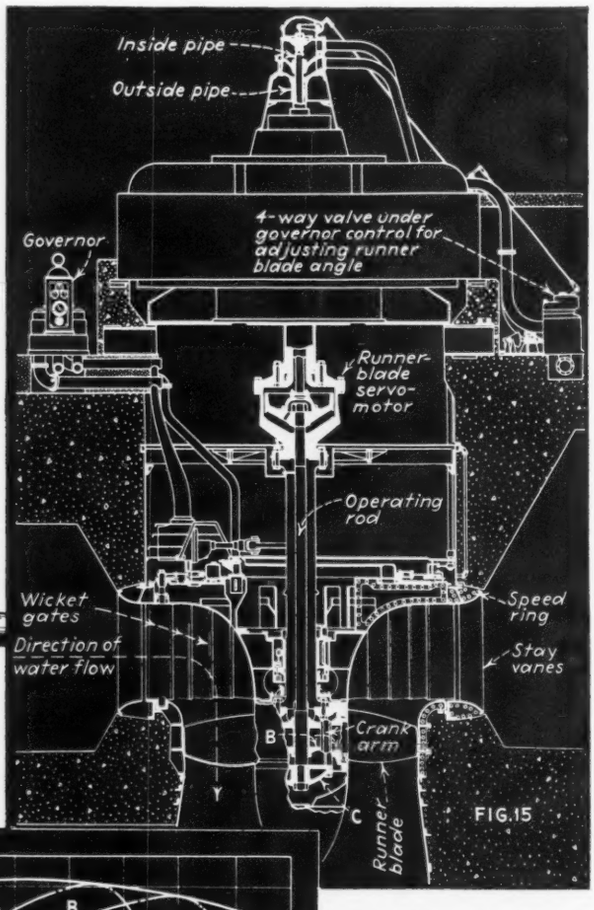
Specific speed, not operating speed, classifies a runner as a high-speed or low-speed type. For example, a 35,000-hp. Francis runner operating under 849-ft. head at 514 r.p.m. is a slow-speed wheel, while a 45,000-hp. propeller wheel running 85.7 r.p.m. under 48-ft. head is a high-speed. To get a true measure of speed, all runner speeds are reduced to specific speed (also called characteristic speed or unit speed), which may be defined as "that speed at which a runner would operate if reduced geometrically to such a size that it would produce 1 hp. under 1 ft. effective head." Its value is given by the formula $N_s = \frac{N\sqrt{HP}}{H^{5/4}}$, in which $N_s$ is specific speed, $N$ is r.p.m., $Hp$. rated horsepower and $H$ rated head in ft.

To "reduce a runner geometrically" means to reduce its dimensions as if they were photographed down. Thus if diameter is reduced three-fourths, all other linear dimensions must be similarly reduced, cutting areas to $\frac{1}{16}$ their original size. On the high-head turbine previously mentioned, the formula gives a specific speed of 21.5 r.p.m., while for the low-head unit it gives 144 r.p.m., more than six times as fast. Practical considerations make it necessary to use a slow-speed wheel for high heads to prevent cavitation, vibration, etc., and a high-speed wheel under low heads to obtain an economical design.

This, in brief, is the story of waterwheel types. How to govern them is told on pages 329 to 332. Subsequent articles discuss other prime movers and their economic applications.

This article is intended to give steam engineers a picture of modern waterwheels and how they operate. No attempt has been made to tell how to select waterpower equipment. If a steam engineer has such a problem he should hire a competent hydro-electric engineer. Development of a waterpower site presents a special problem that can be most economically solved only by engineers who have had considerable experience with this kind of work.



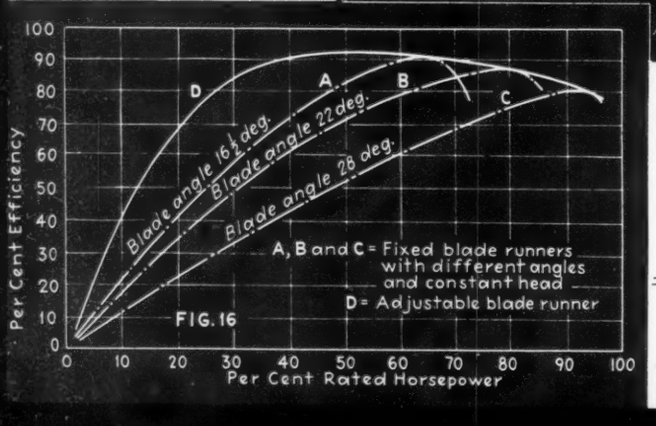




**Fig. 14—Propeller runner. Fig. 15—Cross-secion through Kaplan blade-operating mechanism and runner. Fig. 16—Curves A, B, C, efficiency obtained with propeller turbine under constant head and with fixed blade angles. Curve D, efficiency obtained under constant head, with blade angle adjusted as load changes**

### Some Outstanding Hydro-Electric Units

| Plant Name | Country | No. of Units | Type | Rated Hp. | Effective Head, Ft. |
|---|---|---|---|---|---|
| Dixence..... | Switz. | 3 | I | 42,500 | 5,600 |
| Fulley..... | Switz. | 4 | I | 3,000 | 5,350 |
| Niederenbach. | Switz. | 2 | I | 11,450 | 3,260 |
| Bucks Creek.. | Calif. | 2 | I | 35,000 | 2,360 |
| Big Creek 2A.. | Calif. | 2 | I | 56,000 | 2,290 |
| Balch..... | Calif. | 1 | I | 40,000 | 2,243 |
| Serra..... | Brazil | 2 | I | 40,000 | 2,230 |
| Fusenko..... | Korea | 4 | I | 45,625 | 2,182 |
| Tiger Creek... | Calif. | 2 | I | 36,000 | 1,190 |
| San Francisquito..... | Calif. | 3 | I | 32,000 | 870 |
| Cross Cut.... | Ariz. | 6 | I[1] | 1,000 | 111 |
| Cedar Falls... | Utah | 1 | I[2] | 1,570 | 215 |
| Zappello..... | Italy | 2 | F | 5,000 | 1,152 |
| Bringhausen.. | Germany | 4 | F | 40,500 | 975 |
| Vemork..... | Norway | 1 | F | 17,500 | 918 |
| Oak Grove.... | Ore. | 2 | F | 40,000 | 850 |
| Waterville... | N. C. | 3 | F | 49,000 | 755 |
| Boulder Dam[3] | Nev. | 6 | F | 115,000 | 500 |
| Diablo..... | Wash. | 2 | F | 83,000 | 310 |
| Dnieper..... | U.S.S.R. | 9 | F | 84,000 | 108 |
| Cedar Rapids. | Quebec | 20 | F | 10,800 | 30 |
| Shannon..... | Ireland | 1 | K | 33,000 | 106 |
| Wettingen.... | Switz. | 3 | K | 10,700 | 76 |
| Bonneville[4]... | Wash. | 2 | K | 66,000 | 50 |
| Wheeler Dam. | Tenn. | 2 | P | 45,000 | 48 |
| Safe Harbor... | Penn. | 6 | K | 42,500 | 55 |
| Rock Island... | Wash. | 4 | P[5] | 21,000 | 32 |
| Vargon..... | Sweden | 2 | K | 15,200 | 14 |
| Green Island.. | N. Y. | 4 | P | 3,000 | 13 |
| Rock River... | Ill. | 5 | P | 800 | 8 |

Under "Type," I is impulse, P, propeller; F, Francis; and K, Kaplan.

[1]Vertical-shaft, single-runner type with 6 nozzles. [2]Horizontal-shaft, single-runner design with two nozzles. [3]Seven units being installed, plant laid out for 1,835,000 hp. [4]Two units being installed, plant laid out for 665,000 hp. [5]Blade angle can be adjusted manually while the units are running.

# STEAM TURBINES

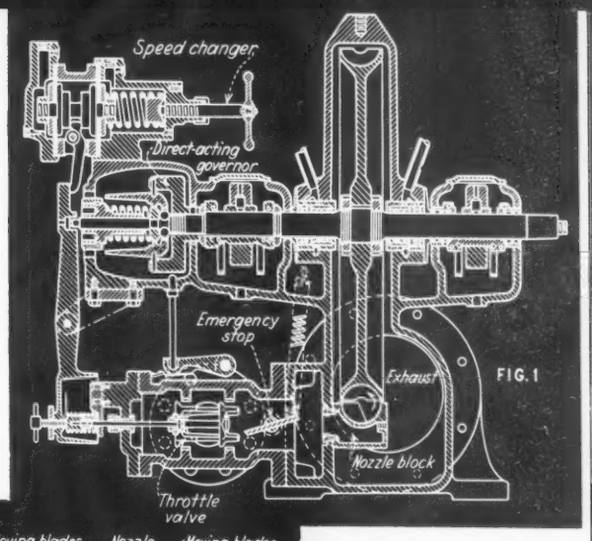


FIG. 1

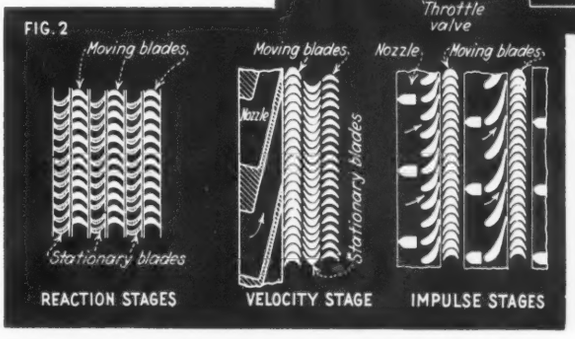



Fig. 1 (Above)—Impulse turbine with helical steam flow. Fig. 2 (Left) — Section through impulse, reaction and velocity stages

TODAY'S steam turbine is the result of gradual improvements made to the original ideas of Carl Gustav De Laval, Charles A. Parsons, Charles A. Curtis and A.C.E. Rateau who, in the latter part of the 19th century, developed the basic turbine types.

The half century that has elapsed has brought the turbine out of the development stage to its present high efficiency, and made it our most important prime mover. Utilities alone have installed steam-turbine drives for electric generators aggregating over 24 million kw., and industry has installed at least one-third of this capacity.

All steam turbines, whether impulse or reaction, convert the kinetic energy of a high-velocity jet of steam into mechanical energy by the impulse action of the jet against the turbine blades and by the reaction of the jet as its direction is changed through about 180 deg. by the blades. Since all turbine rotors receive energy by both impulse and reaction it is fair to ask why some are turbines called "impulse" and some "reaction" types.

In the single-stage impulse turbine, the entire pressure drop and expansion occurs in diverging nozzles. Energy made available by the pressure drop imparts high velocity to the expanding steam. This velocity is absorbed by a single row of turbine blades fastened to the periphery of a disk or wheel. No pressure drop occurs in the blades, but steam velocity is reduced from some high value to a lower value. Such a turbine (originally De Laval) is intrinsically a very high-speed machine, since for best economy the velocity of the moving blades should equal approximately one-half the velocity of the steam as it issues from the nozzles, and steam velocity is high because of the large pressure drop in the nozzles. Steam expanding from 200 lb. abs. to atmospheric in a single diverging nozzle will attain a velocity of about 3,000 ft. per sec.

In a reaction turbine, on the other hand, pressure drop occurs as the steam flows through both the stationary and moving blades. Velocity increases as the steam passes the stationary blading, which forms converging nozzles. The moving blades are usually identical in cross-section. Steam expands in them, thus tends to increase its velocity, but the blade is absorbing velocity at the same time. The net result is a lower leaving velocity than entering velocity. This residual velocity is, however, carried over to the next stationary blade. In reaction turbines, blade velocity is only a little less than the velocity of the steam leaving the stationary blading. But steam velocities are not as high as in impulse turbines because the pressure drop across each stationary and moving blade is kept low by the large number of stages usually employed. (In reaction turbines, a stage is usually considered to be one stationary and one moving row of blades, and in impulse turbines one nozzle and the moving blades up to the next nozzle.) Fig. 2 shows a section through an impulse stage and several reaction stages.

The pressure drop across each row of moving blades in a reaction turbine causes a thrust toward the low-pressure end of the turbine, which in the aggregate is a sizeable force. In small units this is taken care of by thrust bearings developed to withstand considerable loads. Such thrust bearings are usually at the high-pressure end of the turbine and movable so that rotor position may be adjusted. In larger turbines, dummy pistons are provided to balance the thrust developed. In Fig. 3 a cross-section of a mixed-pressure reaction turbine, a dummy piston is indicated at the high-pressure end of the unit. Inlet steam pressure acts against the smaller diameter of the piston, exhaust pressure against the larger area. These areas are calculated so that the difference in pressure acting on them will as nearly as possible balance the thrust produced by the blading. Leakage of steam past the dummy piston is controlled by labyrinth packing.

Another effect of the pressure difference across the moving blades is a tendency for steam to leak around the blades. This leakage is reduced to a minimum by providing small radial and axial clearances. Details of typical construction are shown on page 306.

Reaction-turbine shafts are of large diameter and the diameter changes abruptly, as in Figs. 3 and 4. They are often either drilled or built up to permit inspection.

When pressure or heat drop is large, the blade speed necessary for good effi-

ciency in a single-stage impulse turbine becomes too high, limiting single-stage units to relatively small capacity. So, a little over 40 years ago, Rateau suggested the multi-stage impulse turbine to reduce blade speed (which is limited by the physical properties of available metals to about 1,200 ft. per sec., but more usual velocities are 600 ft. per sec.). By placing a number of nozzle and moving-blade units in series in the same cylinder and on the same shaft, as in Fig. 6, the pressure drop in each nozzle is reduced and velocity of the steam acting against each row of blades is less, hence blade speed need not be so high.

As there is no pressure drop across the blades of an impulse turbine, heavy end thrusts are not encountered and it is not necessary to provide close clearances at the blades. But a seal must be provided between the shaft and the diaphragm that holds the stationary nozzles, for the pressure drop is relatively large. Details of diaphragm shaft seals are shown on page 307.

In impulse turbines, the shaft (compared to that of a reaction unit) is of relatively small diameter. Blades are carried on the periphery of wheels shrunk and keyed to the shaft. This arrangement decreases the leakage area between the diaphragm and shaft and makes a turbine of somewhat cellular construction.

## Velocity Staging

Another method of decreasing required blade speed is velocity staging, invented by Curtis in 1896. In Fig. 2 is a section through a Curtis stage consisting of a diverging nozzle and two rows of moving blades with a stationary blade between them. All the expansion takes place in the nozzle, the intermediate blade merely acting as a guide to redirect the steam into the following row of moving blades. This gives the blading double the opportunity to absorb the energy of the high-velocity steam from the nozzle, and blade velocity need be only about one-quarter the velocity of the steam leaving the nozzle. Fig. 8 shows a 3-pressure-stage turbine, each stage having a Curtis velocity stage. In the first two stages the nozzles do not extend entirely around the periphery of the wheel. Such an arrangement is regularly employed on small units because the volume of steam flowing does not occupy the entire annular area of the blades until it reaches the lower pressure stages.

For various reasons, designers have found it desirable to have a greater heat drop in the first stage than in succeeding stages. This makes velocity in this stage higher than in later ones. Better to utilize this higher velocity, a Curtis velocity stage is often used in the first stage of both impulse and reaction turbines, as in Figs. 4 and 6.

Fig. 6 shows how the blading of an impulse condensing turbine becomes progressively longer as exhaust is approached. In large utility units, exhaust wheels of 12 ft. 9½ in. diameter have been built with blades 38 in. long. To keep wheel diameters and blade lengths within reason, the low-pressure ends of very large turbines are constructed so that steam enters the low-pressure cylinder at the center and flows in either direction toward the ends. With this double-flow arrangement, two sets of blades and nozzles are required, each handling half of the steam flow.

In the turbines discussed so far, steam flow has been axially through the unit. In Fig. 9, the steam flows back and forth through the blades of a single impulse wheel. This re-entry type turbine has one pressure stage and a velocity stage. Units are built with two or more pressure stages and with more than one velocity stage in each pressure stage. Fig. 1 shows another type in which the steam flows in a helix around the periphery of the single wheel. Several nozzle blocks are placed around the wheel, arranged so that steam is directed tangentially against the wheel and is turned back into vanes milled into the nozzle block which in turn redirect the steam into the wheel. Each nozzle block has from three to five directing vanes. Turning vanes in the wheel are also milled in the periphery. It is essentially a multi-velocity-stage impulse turbine, low in first cost, but not very efficient. Both this and the re-entry type are generally confined to auxiliary drives.

The Ljungstrom turbine, used ex-

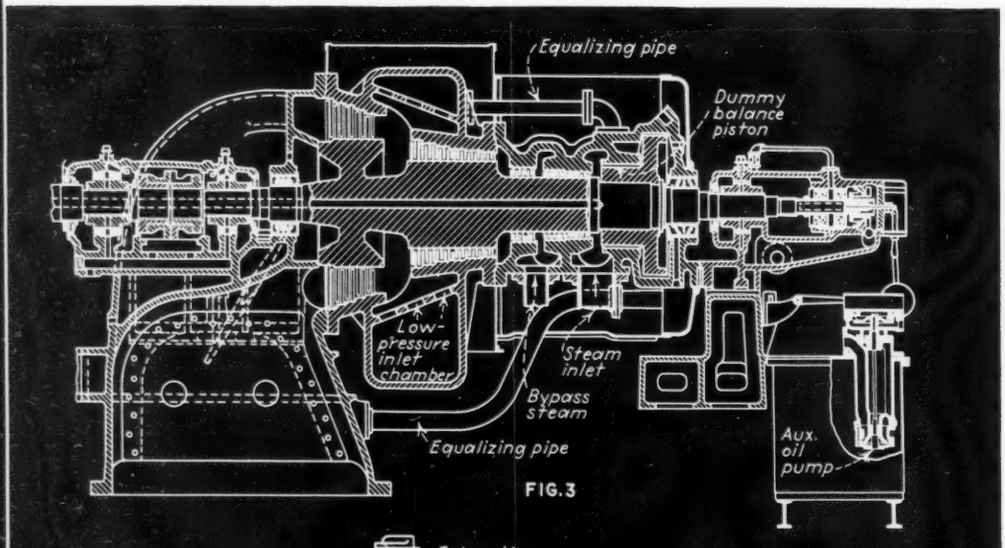


FIG.3

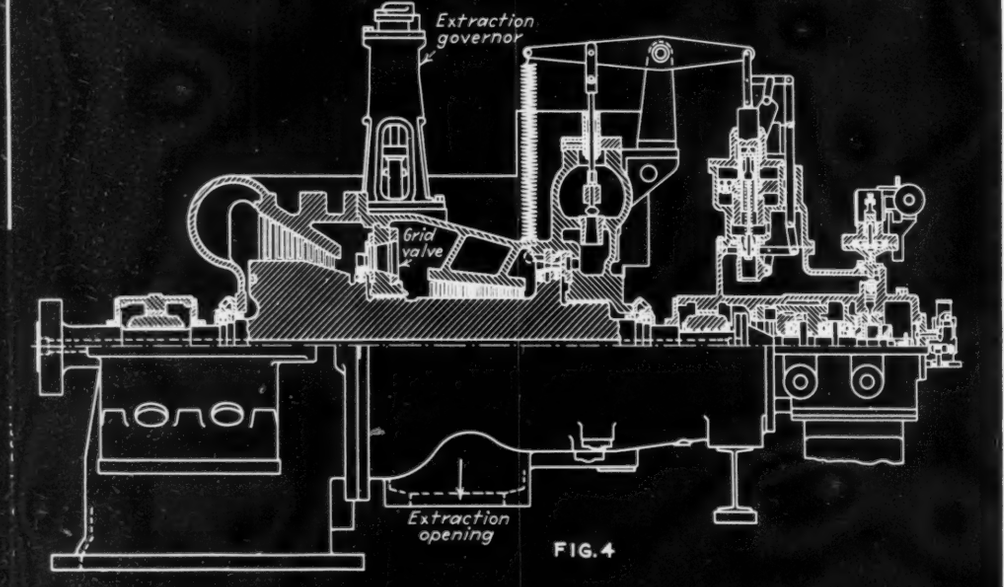


FIG.4

Fig. 3—Mixed-pressure reaction turbine. Fig. 4 (Bottom)—Extraction reaction turbine with Curtis impulse-velocity stage

tensively in Europe but not in America, is a radically different type. Steam enters at the shaft and flows radially outward through concentric rings of reaction blading. There are, however, no stationary blades, one set of blades revolving in one direction and the next row in the opposite direction. This, of course, requires two generators, but even so the turbine is very compact and highly efficient because of the high relative blade speeds.

## Condensing Units

Turbines are more efficient in condensing operation than steam engines and should in general be selected for this service. Figs. 3, 4 and 6 are units designed to run condensing. It is quite common for such turbines to exhaust at 29 to 29.5 in. vacuum. The turbine in Fig. 5 is designed to operate non-condensing, so its exhaust hood and last-wheel blading are relatively smaller than in condensing machines. Unless initial superheat in the steam is high, the exhaust steam at low-pressure will contain a considerably quantity of moisture. This results in blade erosion, a problem that will be treated in "Blading," page 306.

One advantage of the turbine is the ease with which steam can be extracted at any intermediate pressure for heating feedwater or for process. When steam is extracted for feedwater heating only, as in central stations, the pressure of the extracted, or bled, steam (as it is called in this case) depends upon the stage from which it is taken and varies with the load. A unit provided with four bleeder connections is shown in Fig. 6. Such an arrangement would, of course, not be satisfactory when steam at constant pressure is required for process. To extract steam at constant pressure, a grid valve is provided after the stage from which the steam is extracted. When partly closed, it restricts the flow of steam to the lower-pressure stages and thereby increases the pressure at the extraction stage. It is normally under control of a pressure governor which closes the grid when pressure falls and opens it when pressure rises. When electric load on the turbine is light the valve will be nearly closed, but never completely so as some steam must always flow through the low-pressure blading which would otherwise become overheated from friction. The extraction governor is often interconnected with the throttle governor to prevent frequency fluctuation which would otherwise occur when the grid valve opened or closed suddenly. Turbines illustrated in Figs. 4 and 5 are equipped with grid valves and extraction pressure governors.

Some plants have both condensing and non-condensing turbines. The non-condensing units supply process steam, and the condensing unit makes up the deficiency in electric load. In such plants the demand for process steam and power are not always in phase, and exhaust may have to be wasted to the atmosphere. To prevent this, mixed-pressure turbines have been designed so that this excess exhaust can be used in the low-pressure stages of the condensing unit instead of being wasted to the atmosphere. Such a turbine is illustrated in Fig. 3.

Speed of turbines used to drive 60 cycle a.c. generators is fixed at 3,600, 1,800, 1,200 and 900 r.p.m., according to the number of poles in the generator. It can not be more than 3,600 r.p.m., and most industrial units are now designed for this speed. The largest-capacity 3,600-r.p.m. unit in operation is 18,000 kw., but one 50,000-kw. machine is under construction. For capacities below this the tendency is toward high speed. For other drives, the turbine speed may be almost any figure from 1,000 r.p.m. up, depending on size. Frequently units operate at about 5,000 r.p.m. and drive through reduction gears.

Between reaction and impulse turbines there is little upon which to base a choice. Both types can be designed to operate at essentially the same efficiency. They can operate at similar rotative speeds and occupy approximately the same floor space for a given capacity and operating conditions. Extraction, mixed-pressure and back-pressure units with automatic control are available in both types. As far as can be determined, one is as reliable as the other, and they require about equal maintenance. In general, there is little difference in first cost. So the choice comes down to a question of first cost, individual preference, and which manu-

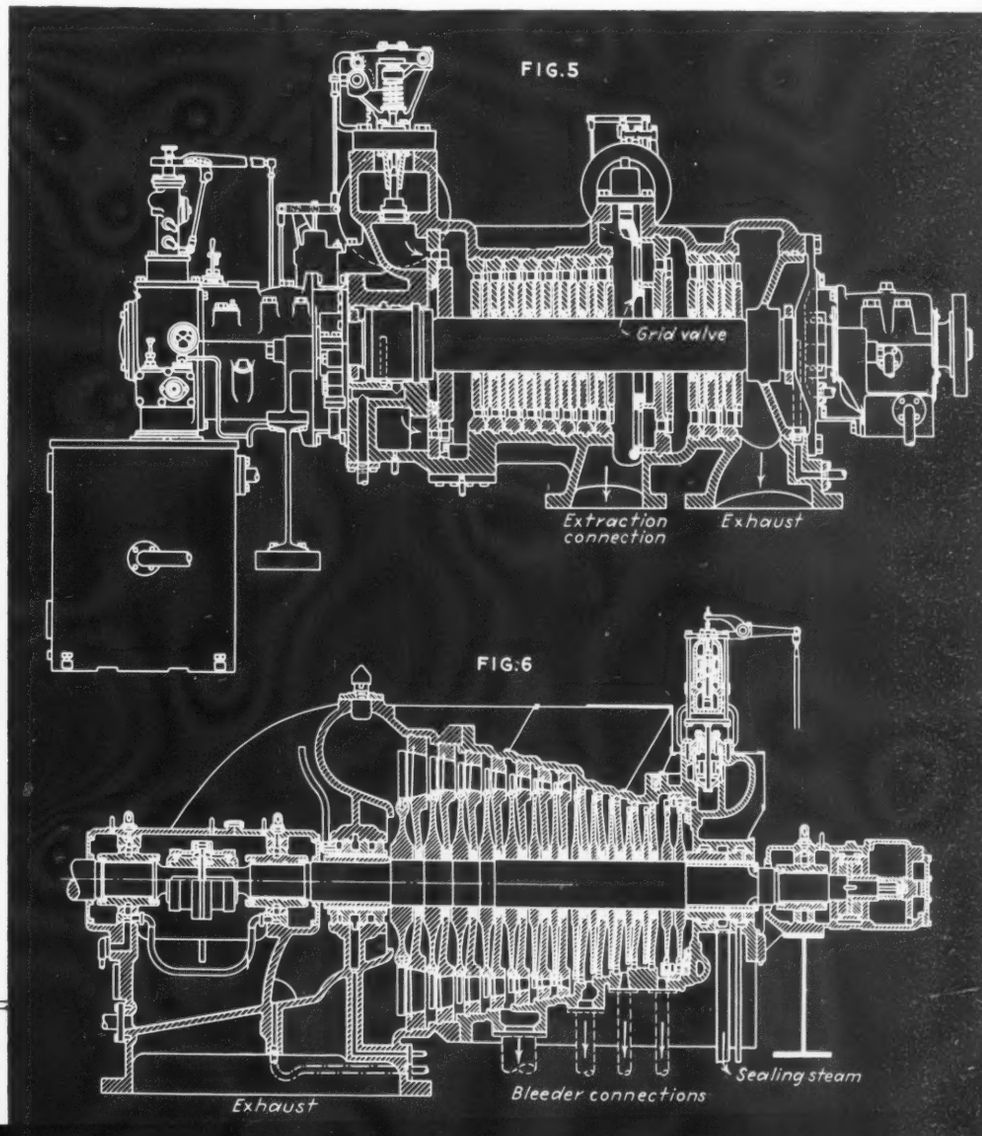



Fig. 5 (Top)—Non-condensing, extraction, impulse turbine. Fig. 6—Impulse turbine with one Curtis stage

facturer designs to meet local requirements most closely. Among smaller auxiliary turbines, there is a greater difference in efficiency which can be used as a factor upon which to base a selection.

Nothing so far has been mentioned about compounding because this article, at least as far as illustrations are concerned, is devoted to industrial types, which are not compounded because their capacity does not warrant such construction. Single-cylinder turbines have been built up to 90,000 kw. Turbines of larger capacity are built with two cylinder, but seldom more than this. 2-cyl. machines with single shaft are tandem compound, the largest being 165,000 kw. 2-cyl. units with two shafts may be cross-compound or vertical-compound. In cross-compound units, the two cylinders are at the same elevation, while in vertical compounding the high-pressure cylinder is placed directly above the low-pressure element. The largest vertical compound unit is 110,000 kw. More than two cylinders are seldom used in America, but 3-cyl. turbines are common in Europe.

In the last stages of turbines, the moisture content of the steam is often high, and with high blade velocities, causes erosion of the blade metal. Considerable effort has been made to remove this moisture as it is formed, so decreasing the erosion of the blading. In general, arrangements for moisture removal have been applied only to large turbines and consist of passages between certain low-pressure stages into which the moisture is thrown by centrifugal force and from which it may be drained to the condenser or low-pressure heater. Tests indicate that from 10 to 30% of the moisture present is removed by such arrangements. It is generally desirable to have not over 10 to 12% of moisture in the exhaust steam.

In the field of high-pressure utility turbines, recent developments include double-case cylinder construction and back-pressure governing. Space between the double-case construction is supplied with exhaust steam at say 200-lb. pressure, thus surrounding the inner high-pressure turbine cylinder and minimizing the pressure differential across the cylinder joint. Back-pressure governing for superposition turbines is being adopted so that the high-pressure unit will follow low-pressure unit loadings and so reduce hunting of the low-pressure unit encountered when load was controlled from the high-pressure end. Likewise, a recent superposition unit has a smaller low-pressure unit in tandem, through which steam for feed heating is taken. Steam flow through it is controlled by an automatic back-pressure governor.

Welding is being used to an increasing extent in turbine construction. Where weight is a factor, as in marine work, turbine casings have been made by forming and welding plate and structural shapes. However, this is not done for stationary turbines. Bedplates for industrial units are being made by welding structural shapes. Nozzle diaphragms are being built by welding plate and bent bar stock. One manufacturer is welding shrouds and lacing wire on to the blading. Oil-piping joints are welded except at joints required to permit dismantling of the piping for inspection and cleaning.

Serious fires have been caused by leakage from oil-system piping coming in contact with superheated-steam piping. To avoid this, piping is rearranged or enclosed in a drain pipe. One manufacturer uses Aroclor, a non-inflammable fluid, in the governing system.

Operating records gathered by the former National Electric Light Assn. show the turbine to be a most reliable prime mover. Of 310 units investigated, average availability has been 96%, including time required for normal overhaul and inspection. Although this figure applies to utility turbines, careful operation, inspection and maintenance should make possible equal performance for the smaller industrial units.

Following articles will discuss details of blade, shaft seals, governing and give steam consumption data.

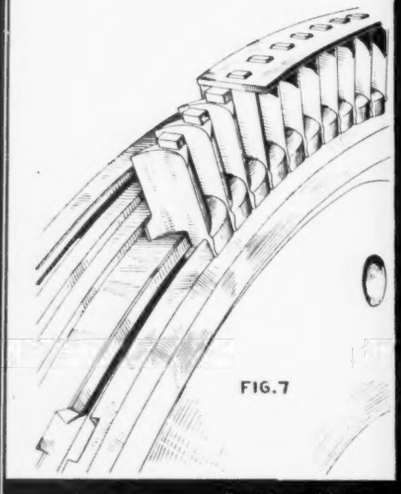


FIG.7

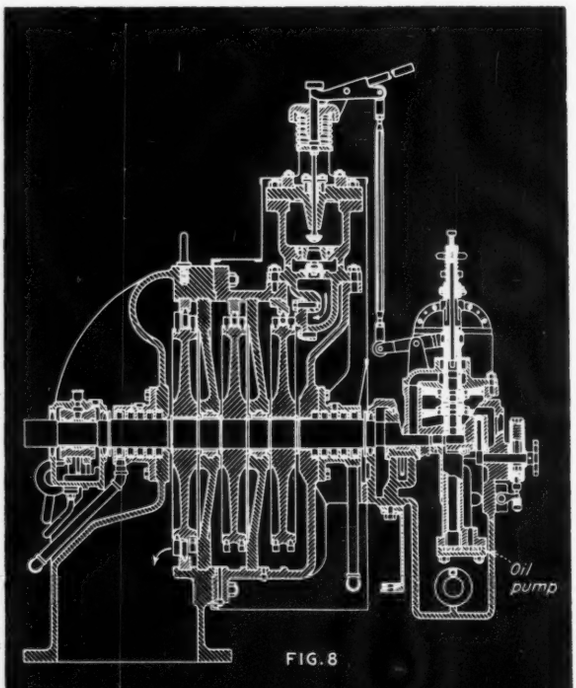



FIG.8

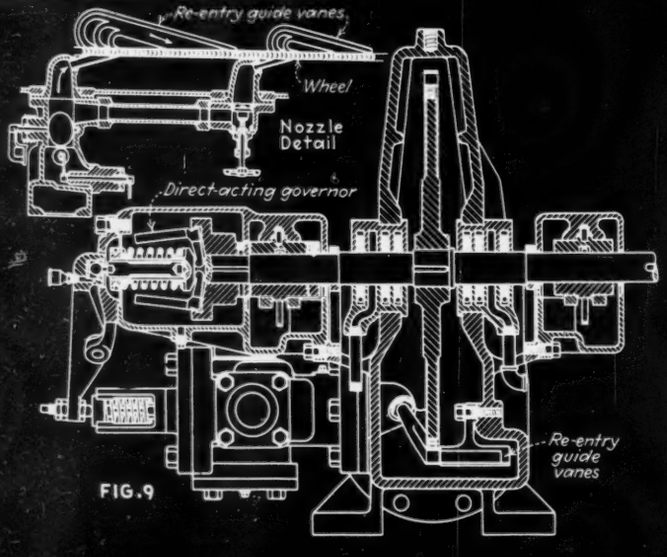



FIG.9

Fig. 7 — Curtis wheel and blading. Fig. 8—Impulse turbine with three Curtis stages

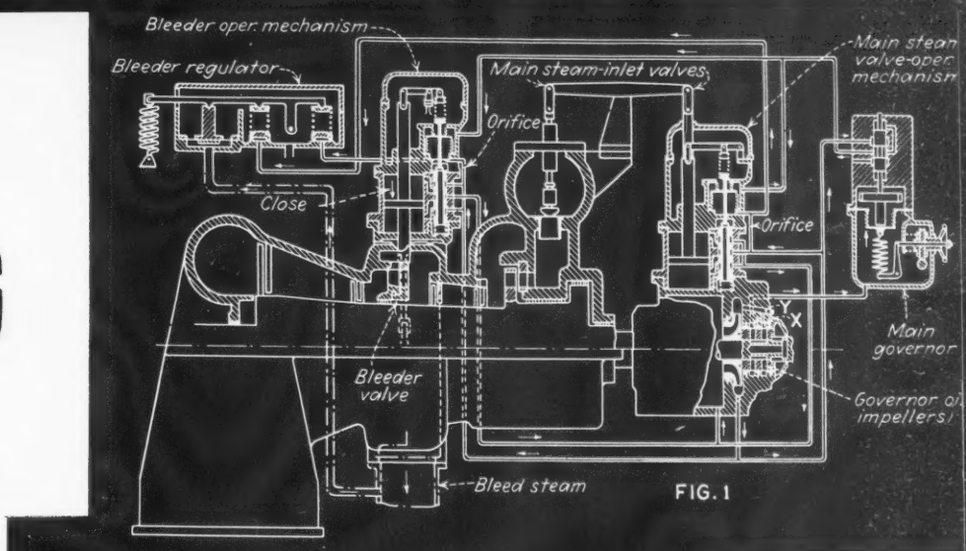
# GOVERNING

LOAD and speed of turbines is controlled either by throttling steam supply to the first stage or by increasing the number of first-stage nozzles in operation to increase load or speed. Both methods are in use for large and small units and for impulse and reaction types. To meet certain conditions both methods are employed, the nozzle control usually being manual and intended to provide overload capacity in emergency. Such capacity is also obtained by bypassing the first stages, admitting steam at initial pressure to a lower-pressure portion of the turbine, as shown in Fig. 3, page 302.

Nozzle governing is the more economical method, but requires more complicated governor mechanisms. For this reason, throttle governing is more frequent in small units. Nozzle governing is usually obtained by placing nozzles in groups and supplying each group through one valve. Valves are arranged to open progressively as load increases, and in this way the velocity in the nozzles and blades is maintained more nearly at the design value.

Small turbines, such as those for auxiliary drive, are direct governed (speed governor operates throttle valve directly). In larger units the control mechanism becomes too large for direct operation, and oil-operated relays and pistons are required to magnify the impulse of the primary governor. Control accuracy likewise requires relay governing. The relay mechanism must be designed so that it responds rapidly to the impulse of the primary governor and without appreciable lag, as discussed in "Governing," page 329.

Primary governors are of either the conventional flyball or the centrifugal oil-impeller type. Fig. 1 illustrates a governing system for a constant-pressure automatic extraction turbine in which the primary governor is of the oil-impeller type. The governor impeller consists of two diametrical holes. Under the impeller, slots are machined in the shaft. A small amount of oil is supplied to chamber *X* from the main oil impeller through an orifice plug, and due to its pressure fills the slots in the shaft. Centrifugal force then delivers oil to annular chamber *Y* and maintains a pressure which varies as the square of the turbine speed. Excess oil drains through center of shaft.



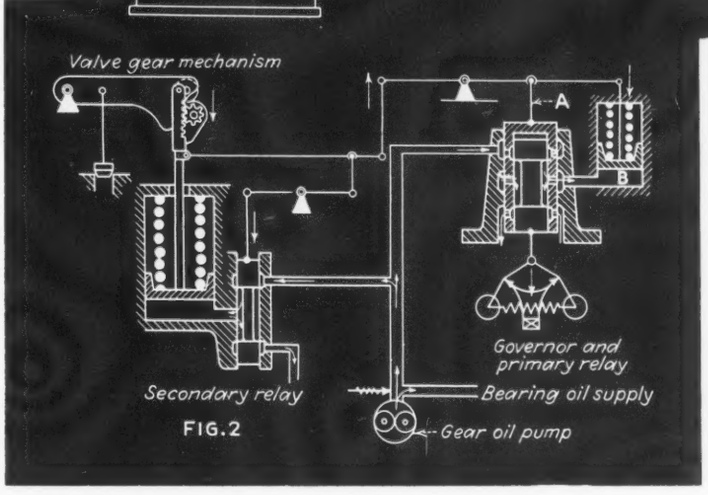




Fig. 1 — Oil governor for automatic-extraction turbine. Fig. 2—Oil relay operated by flyball governor

Oil from the governor impeller exerts pressure against a bellows in the main governor which is opposed by the spring whose tension is regulated by a speed-changing device. An increase in pressure due to a speed increase compresses the bellows which carries with it a relay piston that is kept revolving to reduce friction. This movement causes the upper land of the relay to uncover a port, admitting high-pressure steam to the valve-operating relay and causing it to move to close the throttle. The bleeder-operating relay also receives this increased pressure, causing it to move the grid valve in the closing direction. The increased pressure is also transmitted to the top of the relay. When of sufficient magnitude to balance the pressure increase from the governor impeller, it moves the relay to neutral, thus bringing the system into balance.

The bleeder-pressure regulator operates so that an increase in bleed pressure reduces oil pressure under the sylphon of the main throttle relay, tending to close it and so reduce the steam supply. At the same time it increases the pressure under the sylphon of the grid-valve relay, tending to open it and allow more steam to pass to the condenser. These simultaneous operations tend to keep unit output constant and prevent a speed change with change in demand for process steam.

In large utility units, a secondary non-inflammable fluid called Aroclor is used in the valve-operating pistons.

Fig. 2 shows a flyball governor and relay mechanism for operating nozzle valves. A speed increase causes the flyball governor to pull the relay piston down. Lower lands of the relay uncover ports that permit oil to drain from under piston *B*, which is forced downward by a spring. This movement carries with it the sleeve in the relay, which brings the ports into neutral and also moves the relay of the valve-operating piston so that the piston moves downward. Downward movement of the operating piston operates cams through a rack and pinion, closing the turbine nozzle valves. The piston rod also brings the relay to neutral through a linkage. Speed changes are made by changing length of link *A*.

# BLADING

EXCEPTING time required for regular overhaul and inspection, blade difficulties have caused more turbine outage than any other part of the unit. As a result, manufacturers have spent a great amount of effort on their design and research for materials that will better stand the service requirements.

Illustrations below show various types of both impulse and reaction blading and the methods employed to fasten them to the wheel or spindle. Figs. 2 and 3 illustrate impulse blading from intermediate-pressure stages, one with outside dovetail, the other with inside dovetail. At one point the mating dovetail on the wheel is machined away so the blades can be inserted. Then the blade is held and the wheel rotated until the blade is in place. Fig. 1 shows an entirely different method in a small re-entry turbine. Recesses are milled in the wheel periphery and the blade inserted through a shroud ring. When in place the blade root and wheel are drilled and pinned together.

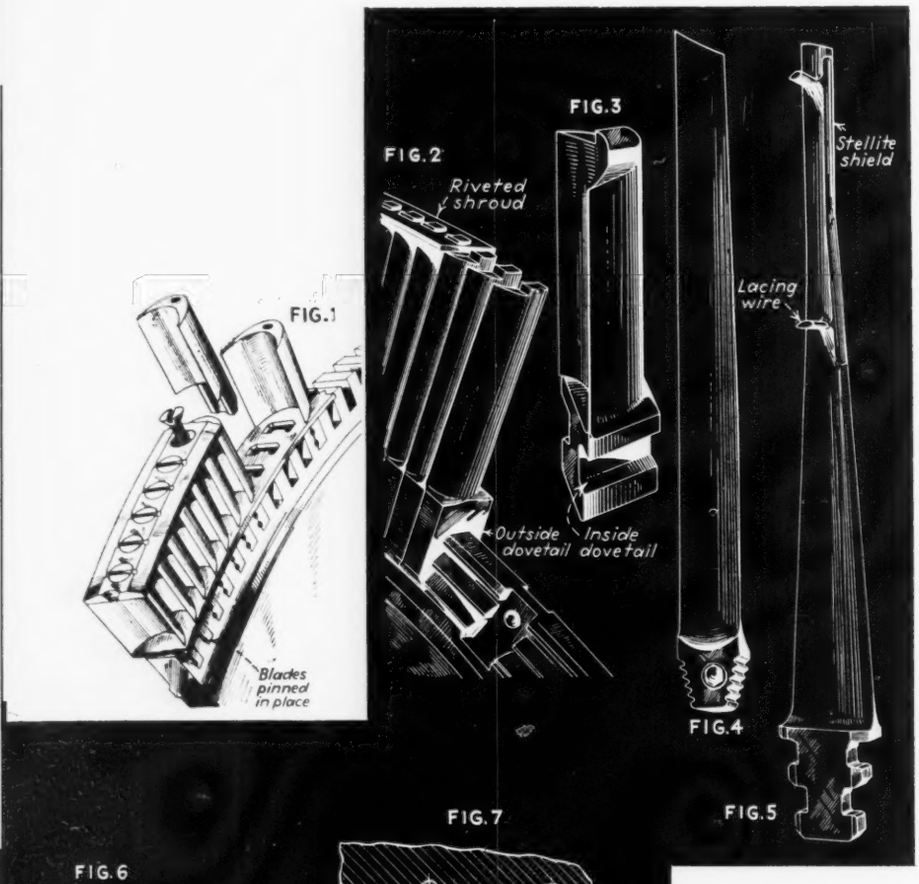
All impulse blades have shrouds at their outer ends to strengthen them against vibration. Shroud rings are usually riveted to the blade, but Fig. 1 shows a shroud held to the blade by screws, and at least one manufacturer is welding the shroud in place.

In reaction turbines, shrouding is frequently not used, particularly for the low-pressure blading. Stiffening of the blades against vibration is then provided by lashing wire. Reaction blades may require both shroud and lashing wire because they have thinner sections than impulse blades. Lashing wire is threaded through holes in the blading and silver-soldered in place. One manufacturer employs a streamlined lashing wire, Fig. 5, to cut down the frictional resistance it introduces, and welds it instead of silver soldering to the blade.

As indicated in "Steam Turbines," page 301, reaction blades must have close clearances to prevent leakage of steam past the blades. Fig. 6 shows the sealing arrangement in one reaction turbine. Radial seal is provided by thin strips of special alloy rolled into grooves cut in the cylinder and rotor. The strip is sufficiently thin so that any rubbing will wear the strip away without causing local heating. Clearances range between 0.02 and 0.1 in.

High-pressure and intermediate-pressure blades have the same width throughout their length. Low-pressure blades for large-capacity units are wider at the root than at the blade tip, as in Fig. 5, which also shows the blade twisted or warped so that the entrance angle will be proper for the higher blade velocity at the blade tip. Long blades are usually highly stressed, and arrangements for holding them on to the wheel or rotor must be stronger, hence the double dovetail and serrations of Figs. 4 and 5.

Steam in the low-pressure stages of condensing turbines frequently has 10 to 12% moisture. With high blade velocity, moisture causes considerable erosion of blade material. Much research work has been done to find metals that will withstand this erosive action, which occurs mostly at the entrance edge of the blade. Stainless steels are being used where this problem is encountered, and when conditions are very severe, Stellite shields are welded on to protect the back of the inlet edge. Stainless steel and nickel steels are used for high-pressure blades when temperatures are high. Various copper alloys are satisfactory for reaction blades when temperatures are not high.



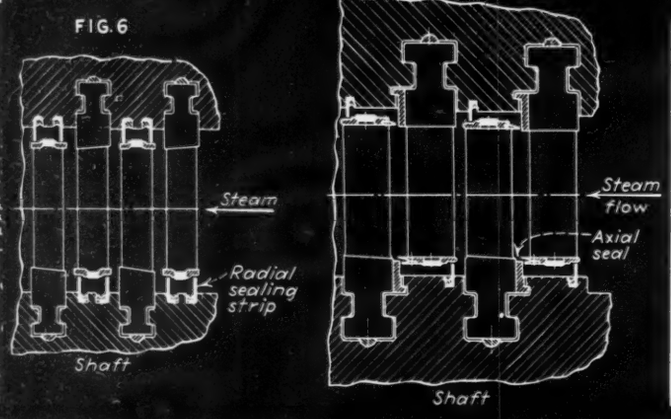




Fig. 1—Impulse blade, drilled and pinned. Fig. 2 — Impulse blade with outside dovetail. Fig. 3—Impulse blade with inside dovetail. Fig. 4—Low-pressure reaction blade. Fig. 5—Low-pressure reaction blade, showing lacing wire. Figs. 6 and 7—Reaction blades with clearance detail

# GLANDS & SEALS

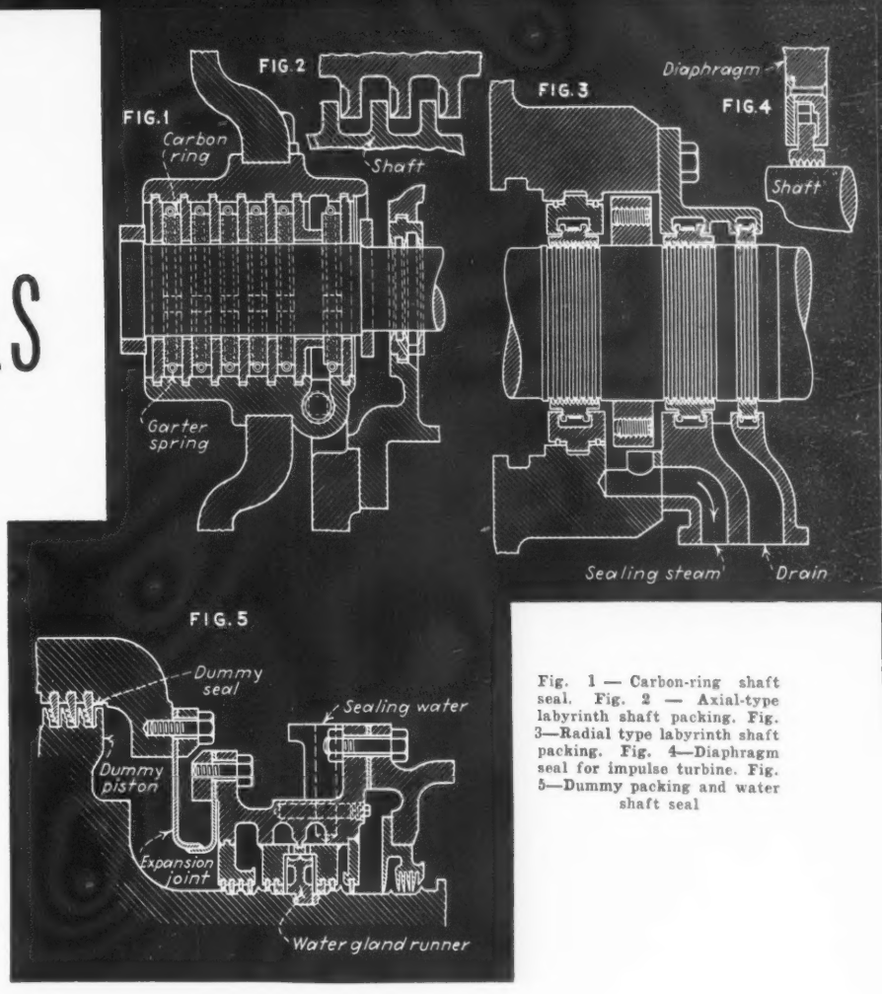



Fig. 1 — Carbon-ring shaft seal. Fig. 2 — Axial-type labyrinth shaft packing. Fig. 3—Radial type labyrinth shaft packing. Fig. 4—Diaphragm seal for impulse turbine. Fig. 5—Dummy packing and water shaft seal

SEVERAL places in turbines must be sealed to prevent leakage of steam. Seals must be provided where the shaft goes through the turbine casing to prevent steam leaking out at the high-pressure end, and in condensing units to prevent air from leaking in at the exhaust end. In impulse turbines, seals are provided at the shaft for each nozzle diaphragm to prevent steam leaking from one stage to the next lower pressure stage. In reaction turbines, a seal is necessary for each moving and stationary blade, as indicated on page 306. When dummy balancing pistons are used, they must be provided with seals to prevent leakage of steam from the high pressure side to the low pressure side.

Three general types of seals are in use: labyrinth, water, and carbon ring packing. Many variations exist, so only a few typical examples are shown.

Fig. 2 shows a simple radial-form labyrinth shaft packing. A number of rectangular grooves are machined in the shaft in three groups where it passes through the cylinder. Rings with knife-edge projections, machined to correspond to both grooves and raised portion between grooves, are held in the seal housing. Clearance between knife-edge and shaft is only a few thousandths of an inch. Steam leaking through this small clearance must follow a very tortuous path that offers considerable resistance. An intermediate steam leak-off connection is provided, likewise a drain for condensate, followed by a smaller labyrinth. Leak-off pressure is usually from 3 to 10 lb. Some radial labyrinths have knife-edge projections extending from both the shaft and the stationary seal. One manufacturer makes these projections of thin alloy steel ribbon rolled into grooves, as in Fig. 7, page 306.

An axial type seal, Fig. 3 has a shaft sleeve with a number of collars machined on it. The packing ring, with an equal number of L-shaped rings, is placed so the L-shaped projections are between the collars. The angle projections of the packing ring are thinned down to a narrow face, and a small clearance maintained between this face and the collars.

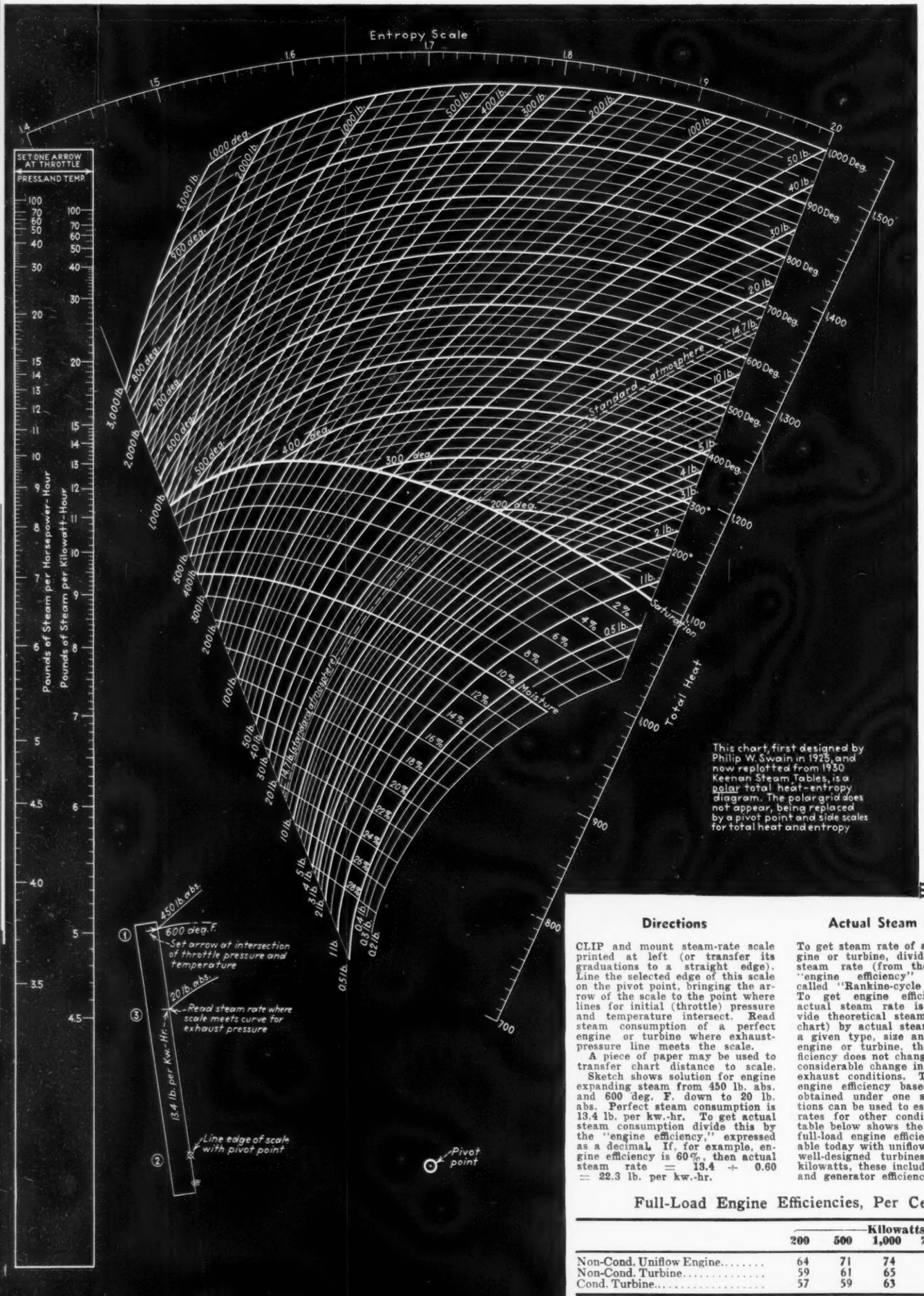
A combination labyrinth and water seal and labyrinth dummy packing for a reaction turbine is illustrated in Fig. 5. The dummy packing has small clearances both radially and axially, and the projecting fingers are longer than in shaft seals. The shaft-gland casing is bolted to the bearing and connected to cylinders by a welded-steel expansion joint. With this arrangement, the gland can be dismantled without lifting the turbine-cylinder cover. The water runner is made in halves, set in a groove in the turbine shaft and caulked in place. Water is supplied to the runner through holes drilled through from the supply pocket. When the turbine comes up to speed, the runner maintains a film of water at its periphery which prevents leakage of air or steam. Labyrinth packing on either side prevents steam from leaking past and out of the runner casing and also prevents steam from leaking in to the runner when starting. This is a high-pressure-end seal, but exhaust-end seals are similar.

Diaphragms are usually sealed with radial glands, as in Fig. 4. This gland ring is made in four parts, each held in place by flat springs. Frequently a shaft sleeve is provided to take wear that might occur in case the seal rubs.

Industrial turbines are frequently provided with carbon-ring packing, as in Fig. 1. This packing has six carbon rings, each about $\frac{1}{2}$ in. wide, carefully fitted to the shaft and held between partitions. Each ring is in three segments held against the shaft by a garter spring that encircles the carbon ring. A condensate drain is provided between the 6th and 5th rings. Sometimes flat springs are used instead of garter springs and the rings divided into four instead of three segments. Greater or smaller numbers of rings are used, depending on the pressure to be sealed and the shaft diameter. Performance of carbon-ring seals depends on smooth polished surfaces on both ring and shaft. If either surface is scored, excess leakage is likely.

In large units the leak-off from steam seals often amounts to a considerable quantity and is frequently used for feedwater heating. Steam leak-off from high-pressure steam seals is highly superheated due to the constant-heat pressure reduction through the seal.

# STEAM-RATE CHART




## Directions

CLIP and mount steam-rate scale printed at left (or transfer its graduations to a straight edge). Line the selected edge of this scale on the pivot point, bringing the arrow of the scale to the point where lines for initial (throttle) pressure and temperature intersect. Read steam consumption of a perfect engine or turbine where exhaust-pressure line meets the scale.

A piece of paper may be used to transfer chart distance to scale.

Sketch shows solution for engine expanding steam from 450 lb. abs. and 600 deg. F. down to 20 lb. abs. Perfect steam consumption is 13.4 lb. per kw.-hr. To get actual steam consumption divide this by the "engine efficiency," expressed as a decimal. If, for example, engine efficiency is 60%, then actual steam rate = 13.4 ÷ 0.60 = 22.3 lb. per kw.-hr.

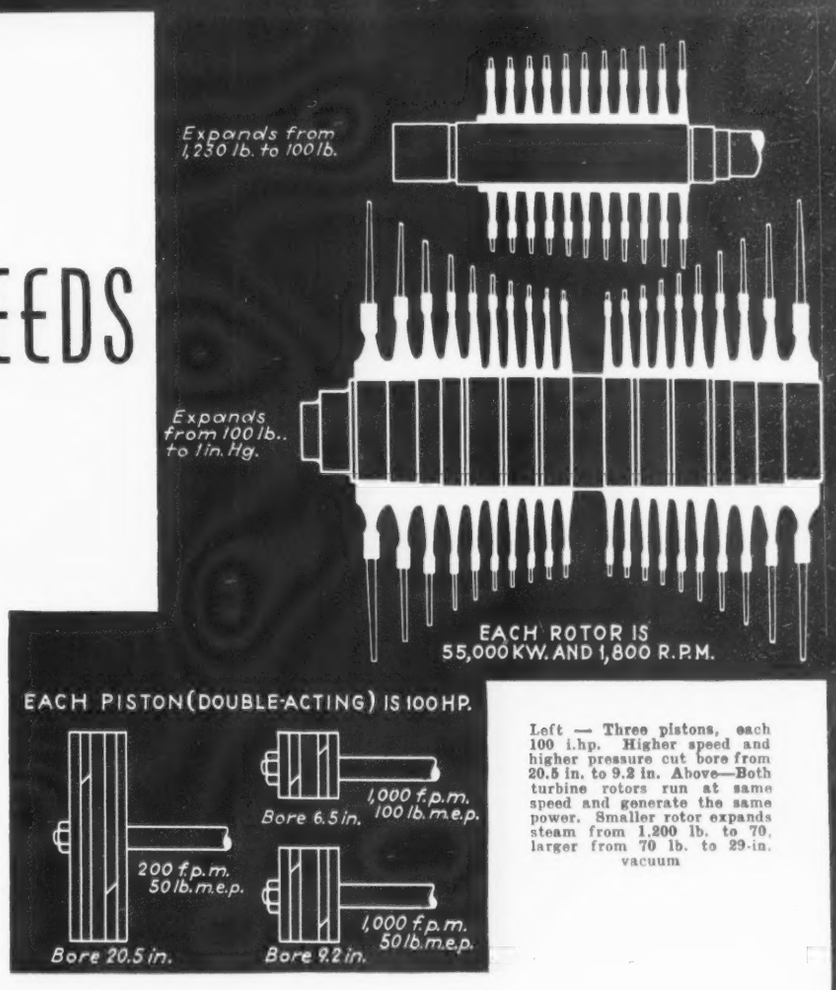
## Actual Steam Rates

To get steam rate of an actual engine or turbine, divide theoretical steam rate (from the chart) by "engine efficiency" (sometimes called "Rankine-cycle efficiency"). To get engine efficiency where actual steam rate is known, divide theoretical steam rate (from chart) by actual steam rate. For a given type, size and loading of engine or turbine, the engine efficiency does not change much with considerable change in throttle and exhaust conditions. Therefore the engine efficiency based on results obtained under one set of conditions can be used to estimate steam rates for other conditions. The table below shows the approximate full-load engine efficiencies obtainable today with uniflow engines and well-designed turbines. Based on kilowatts, these include mechanical and generator efficiencies.

### Full-Load Engine Efficiencies, Per Cent

| | Kilowatts | | | | |
|---|---|---|---|---|---|
| | 200 | 500 | 1,000 | 2,000 | 5,000 |
| Non-Cond. Uniflow Engine | 64 | 71 | 74 | .. | .. |
| Non-Cond. Turbine | 59 | 61 | 65 | 70 | 75 |
| Cond. Turbine | 57 | 59 | 63 | 68 | 73 |

# PRESSURES—SPEEDS


Expands from 1,230 lb. to 100 lb.
Expands from 100 lb.. to 1 in. Hg.
EACH ROTOR IS 55,000 KW. AND 1,800 R.P.M.
EACH PISTON (DOUBLE-ACTING) IS 100 HP.
1,000 f.p.m. 100 lb. m.e.p.
Bore 6.5 in.
200 f.p.m. 50 lb. m.e.p.
1,000 f.p.m. 50 lb. m.e.p.
Bore 20.5 in.
Bore 9.2 in.

Left — Three pistons, each 100 i.hp. Higher speed and higher pressure cut bore from 20.5 in. to 9.2 in. Above—Both turbine rotors run at same speed and generate the same power. Smaller rotor expands steam from 1,200 lb. to 70, larger from 70 lb. to 29-in. vacuum

IN ALL prime movers higher speed means fewer pounds of metal per horsepower (also fewer pounds per kilowatt, both in the generator and in the generating set). It also means less floor space, less cubical space, lighter foundations. Within limits, it means lower first cost.

In general, higher speed means less reliability and higher maintenance, although this effect may be hardly noticeable up to a fairly high speed. The practical speed limits of all types of prime movers are constantly rising as metals and designs improve with experience and research.

Since 1 hp. equals 33,000 ft. lb. per min., a double-acting steam engine, with an m.e.p. of 33 lb., a piston area of 100 sq.in., and a piston speed of 10 ft. per min., will generate 1 hp. (theoretically). At 100 ft. per min. it will generate 10 hp.; at 1,000 ft. per min., 100 hp. At 10,000 ft. per min. (if that were possible) it would generate 1,000 hp.

With a purely rotating machine, such as a steam turbine, precise balancing keeps the shaking forces within control. Aside from building in enough stiffness to keep the critical speed well above the operating, the high-speed problem in turbine design is mainly one of scientifically shaping wheels and blades and using alloys capable of withstanding the very high centrifugal forces involved. In this field we see the present end point in 1,000-hp. turbines running at 20,000 r.p.m. and weighing no more than 5 oz. per hp.

The imaginary steam engine discussed a while back shows how power goes up with speed. Here is a similar comparison for the steam turbine. Suppose the tangential force (figured at the blade centers) amounts to 330 lb. Then if points at this radius move 100 ft. per min., the turbine will deliver 1 hp. Theoretically, at 1,000 ft. per min., it will deliver 10 hp.; at 10,000 ft., 100 hp.; at 100,000 ft., 1,000 hp.; and so on. In practice, of course, the problem is much more complicated than this, because the force on the turbine blades depends on the blade velocity as well as that of the jet.

The speed of a steam turbine, then, is a compromise not only with strength of metals and refinements of design, but also with the steam conditions, number of stages desired and many other conditions.

The user does not need to go into these matters in detail. Presumably the builder will make any cost saving he can by raising speed so long as it does not endanger safety or increase maintenance. The part of the user is to be aware of the fundamental dollar-saving attributes of higher speed and not let some old-fashioned notion of "how fast is too fast" restrict his choice to his own disadvantage. He must lean rather on the experience, reputation and guarantees of an established manufacturer.

The same applies in the case of steam engines. Here higher speeds bring great savings in weight, but with them new problems of design and metals of moving parts—also problems relating to rapid flow of steam through valves and ports.

Raising the steam pressure always reduces the size of a steam engine or turbine required to generate a given power. In the case of a steam turbine, raising the back pressure also greatly reduces the size. In this case the steam is dense throughout the turbine and only small blade passages are required. Probably the most compact power units of any sort ever developed are steam turbines running at 10,000 r.p.m. or more, taking steam at 1,400 lb. pressure and exhausting at several hundred pounds pressure.

The relatively small size of the high-pressure end or high-pressure cylinder of power units has been a distinct factor in promoting the use of higher steam pressures. In one respect, though, this small size is a restriction. For small powers the blades of high-pressure turbines become excessively short, wasting much steam through the clearances. For this reason it practically never pays to install a 100 or 200-hp. turbine for 1,400-lb. pressure. This opens a new field for the reciprocating engine as a small high-pressure unit.

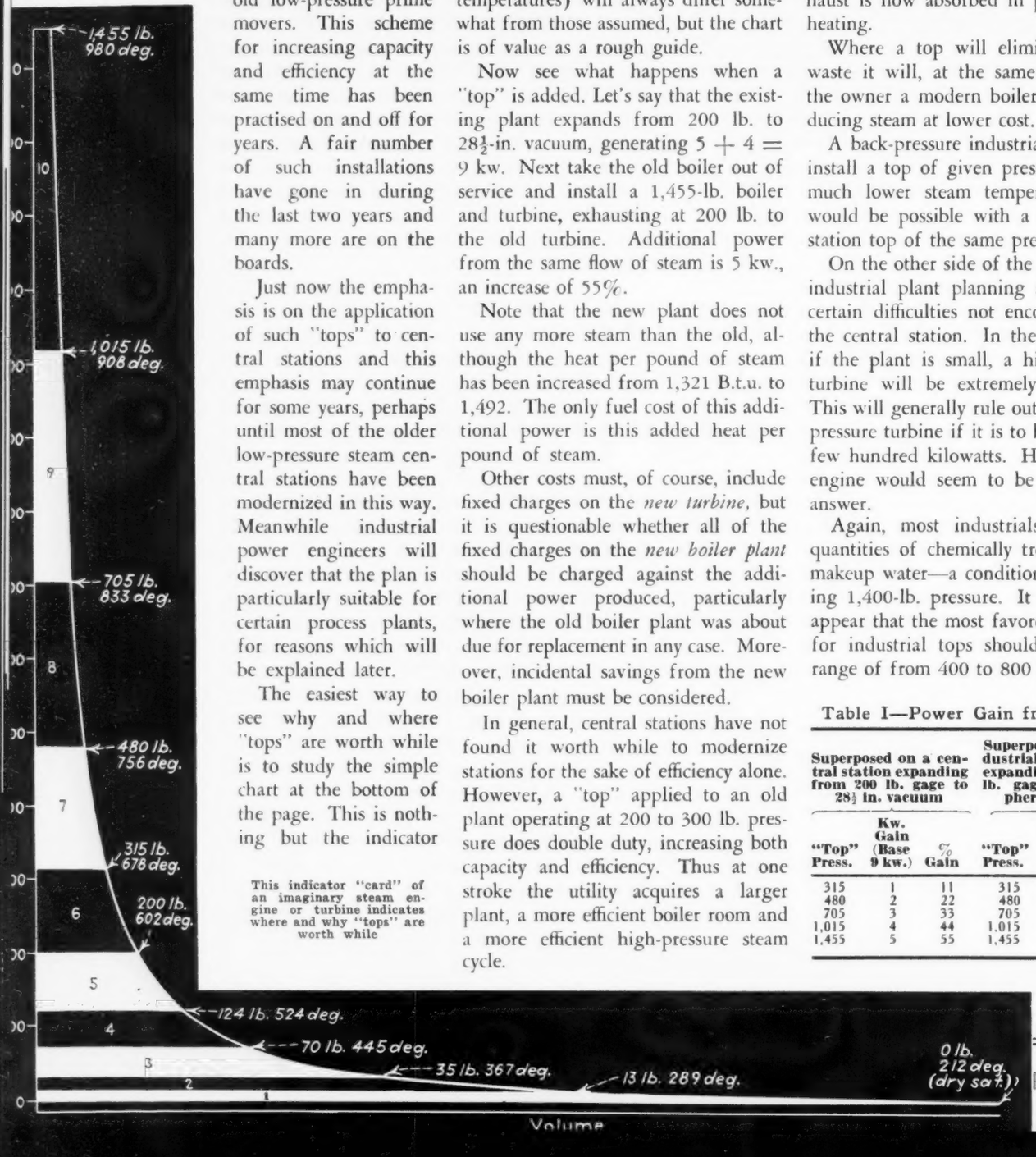
Of course, the primary reason for raising steam pressures has not been to reduce the size of prime movers, but rather to obtain a greater heat drop and thus get more power out of a given flow of steam. This matter is discussed further in "Tops," next page.

# "TOPS"

A "TOP," power slang for "superposed plant," is a high-pressure steam power plant whose turbines or engines exhaust into lower-pressure engines or turbines. Supposedly, the whole combination might be built at one time, but a "top" today is almost always a new high-pressure plant superposed on old low-pressure prime movers. This scheme for increasing capacity and efficiency at the same time has been practised on and off for years. A fair number of such installations have gone in during the last two years and many more are on the boards.

Just now the emphasis is on the application of such "tops" to central stations and this emphasis may continue for some years, perhaps until most of the older low-pressure steam central stations have been modernized in this way. Meanwhile industrial power engineers will discover that the plan is particularly suitable for certain process plants, for reasons which will be explained later.

The easiest way to see why and where "tops" are worth while is to study the simple chart at the bottom of the page. This is nothing but the indicator "card" of an imaginary steam engine or turbine.




This indicator "card" of an imaginary steam engine or turbine indicates where and why "tops" are worth while

This machine passes exactly 100 lb. of steam per hour. Each white or black area on the diagram represents an output of one kilowatt from this flow of steam. Thus, if expansion is from 480 lb. gage to 35 lb. gage, the "card" consists of the five areas: 3, 4, 5, 6, 7, and the output per 100 lb. of steam per hr. is 5 kw. (20 lb. steam per kw.-hr.). If expansion is from 1,455 lb. gage to atmospheric, the output per 100 lb. per hr. is 10 kw., and so on. Expansion down to 28½-in. vacuum would add 4 kw. more.

In plotting this chart it was assumed that each stage (each area) operates with 70% engine efficiency. Steam temperatures were chosen to give dry saturated steam at atmospheric pressure. Actual conditions (particularly temperatures) will always differ somewhat from those assumed, but the chart is of value as a rough guide.

Now see what happens when a "top" is added. Let's say that the existing plant expands from 200 lb. to 28½-in. vacuum, generating 5 + 4 = 9 kw. Next take the old boiler out of service and install a 1,455-lb. boiler and turbine, exhausting at 200 lb. to the old turbine. Additional power from the same flow of steam is 5 kw., an increase of 55%.

Note that the new plant does not use any more steam than the old, although the heat per pound of steam has been increased from 1,321 B.t.u. to 1,492. The only fuel cost of this additional power is this added heat per pound of steam.

Other costs must, of course, include fixed charges on the *new turbine*, but it is questionable whether all of the fixed charges on the *new boiler plant* should be charged against the additional power produced, particularly where the old boiler plant was about due for replacement in any case. Moreover, incidental savings from the new boiler plant must be considered.

In general, central stations have not found it worth while to modernize stations for the sake of efficiency alone. However, a "top" applied to an old plant operating at 200 to 300 lb. pressure does double duty, increasing both capacity and efficiency. Thus at one stroke the utility acquires a larger plant, a more efficient boiler room and a more efficient high-pressure steam cycle.

It is not practicable here to consider all these factors, nor such matters as the effect of top pressure on bleeder heating, air heating, etc. These are problems of a specific case. So let's consider only a constant flow of steam and see how a "top" affects the power gain under various conditions.

The table is based on the chart. Note that the back-pressure plant shows nearly twice the percentage gain in power from a given rise in pressure.

Expressed as a rule, one can say that low initial pressure and high back-pressure both favor the addition of a top *if* it is desirable to get more power from a given flow of steam. The *"if"* here is very important. It means that process industrials now sending much steam to atmosphere or condenser can save a lot of fuel with a top, but can save no fuel whatever if all the exhaust is now absorbed in process and heating.

Where a top will eliminate steam waste it will, at the same time, give the owner a modern boiler room producing steam at lower cost.

A back-pressure industrial plant can install a top of given pressure with a much lower steam temperature than would be possible with a condensing station top of the same pressure.

On the other side of the picture, the industrial plant planning a top faces certain difficulties not encountered by the central station. In the first place, if the plant is small, a high-pressure turbine will be extremely inefficient. This will generally rule out a 1,400 lb. pressure turbine if it is to be of only a few hundred kilowatts. Here a steam engine would seem to be the logical answer.

Again, most industrials use large quantities of chemically treated boiler makeup water—a condition not favoring 1,400-lb. pressure. It would thus appear that the most favored pressures for industrial tops should lie in the range of from 400 to 800 lb.

**Table I—Power Gain from "Top"**

| Superposed on a central station expanding from 200 lb. gage to 28½ in. vacuum | | | Superposed on an industrial power plant expanding from 200 lb. gage to atmospheric pressure | | |
|---|---|---|---|---|---|
| "Top" Press. | Kw. Gain (Base 9 kw.) | % Gain | "Top" Press. | Kw. Gain (Base 5 kw.) | % Gain |
| 315 | 1 | 11 | 315 | 1 | 20 |
| 480 | 2 | 22 | 480 | 2 | 40 |
| 705 | 3 | 33 | 705 | 3 | 60 |
| 1,015 | 4 | 44 | 1,015 | 4 | 80 |
| 1,455 | 5 | 55 | 1,455 | 5 | 100 |

# STEAM ENGINES

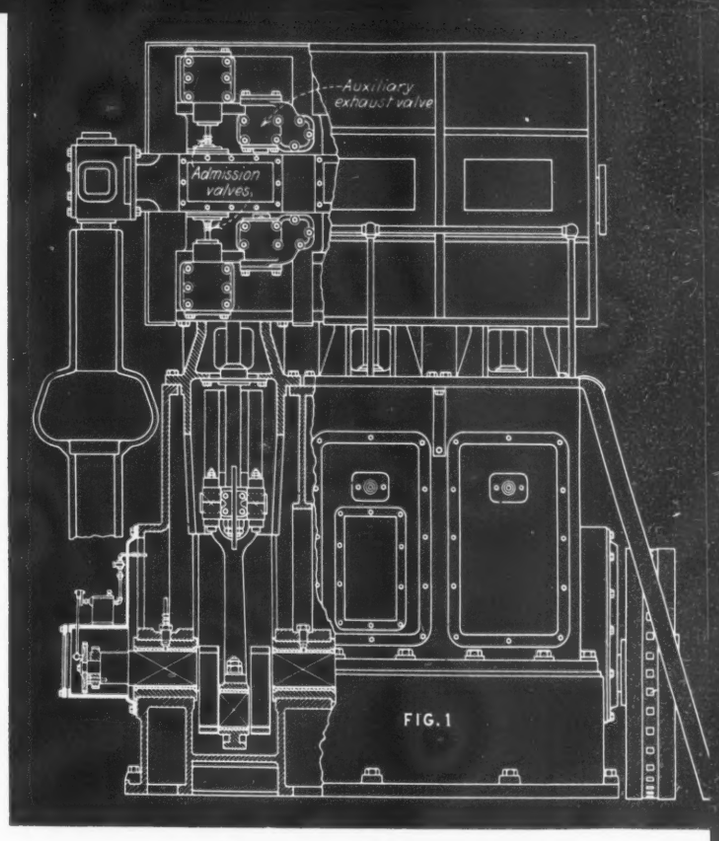



Fig. 1—4-cyl. vertical non-condensing uniflow engine with auxiliary exhaust valves

TWO thousand and more years ago, Hero of Alexandria described steam-using devices. Among them were an engine used to open and close temple doors and the ancestor of the turbine. After fifteen hundred years, other men began to think that there might be something to this peculiar vapor that arose when water was heated. They put it to work, and evolved the steam engine. That brings us up to a story now well known—the tale of the work of Newcomen, Watt, Smeaton, Cartwright, Hornblower, Corliss, Evans, and many another, a saga of development that made the steam engine the basic prime mover fifty years ago.

Engines have long passed the development stage and little change of major proportion has been made for many years. It is in details and in applications that the development has been made, in use of better materials and in refinements in valve operation, lubrication, governing, etc. These changes, singly unimpressive, have together made the steam engine worthy of consideration in most industrial power plants.

Counterflow, single-valve engines, a simple type, are lower in first cost and not as efficient as other types. Vertical units, like that shown in Fig. 2, are used quite commonly for auxiliary drives. Their inefficiency there is unimportant, if their exhaust can be fully recovered in feedwater heating or process without displacing other exhaust steam. Engines of this type are built in sizes up to about 400 hp., and for maximum speeds of 600 to 700 r.p.m. Pressures are seldom over 175 lb. per sq.in. They are not usually classed as prime movers.

Uniflow and 4-valve engines are generally used as prime movers for driving generators. Capacities range up to 1,500 or 2,000 kw., with a few installations larger than that, and steam pressures run 150 to 350 lb. High temperatures introduce difficulties with cylinder and valve lubrication. But two single-acting compound engines have been built for 1,500 lb. and 780 deg.

Fig. 3 shows a single 4-valve engine with poppet valves operated by cams driven by eccentrics on a gear-driven layshaft. Separate eccentrics are provided for each valve so that admission and cutoff events may be controlled separately from exhaust events, thus permitting much earlier cutoff than that obtainable with single-valve engines. These engines are therefore more efficient at part loads. The governor is on the lay shaft and controls engine speed by changing throw and angular relation of the eccentrics operating admission valves on top of the engine cylinder. Four-valve engines are also built with oscillating cylindrical or drop-piston valves operated by Corliss, cam, or toggle mechanisms, as explained in "Valves & Gears."

Cylinder condensation due to the cooling effect of the expanded exhaust steam is one loss in a steam engine. To decrease it, compounding was formerly employed; that is, steam was expanded in two or more cylinders instead of one, thus decreasing the temperature difference between inlet and exhaust in each cylinder. In stationary practice, compounding is now seldom used unless pressures are very high or the engine exceptionally large.

In place of compounding, the uniflow engine has come into use. It has exhaust ports at the center of the cylinder, and the relatively longer piston acts as the exhaust valve, uncovering the central ports just before reaching the end of its stroke. With this arrangement, the middle portion of the cylinder is relatively cool and the exhaust does not pick up heat from the cylinder as it does when exhaust is through the hot end of a counterflow engine. Likewise compression occurs very early, and the resulting high terminal temperature aids in preventing heat transfer and condensation during admission. This early compression, while satisfactory for condensing operation, will cause too high a terminal pressure when operating non-condensing unless large clearance volume or auxiliary exhaust valves are provided. Since clearance volume is detrimental to economy, auxiliary exhaust valves are employed in engines intended for non-condensing operation as in the horizontal uniflow engine of Fig. 4.

Uniflow engines usually have lower mean effective pressures because of early compression and cutoff. Hence, under similar operating conditions a larger-diameter cylinder or longer stroke is required for a given power capacity than for counterflow engines. Cylinders of uniflow engines are longer than those of counterflow engines of the same stroke because piston length is about 90% of the stroke. For these reasons, the uniflow usually costs more than counterflow types but less than

compound engines, which it about equals in efficiency.

Horizontal uniflows are usually provided with tail rods. The crosshead and tail-rod bearings are adjusted so that only the piston rings touch the cylinder, thus eliminating the wear and friction that would otherwise be caused by the heavier and longer uniflow piston.

Both horizontal and vertical uniflows are in regular use. Vertical units usually have two or more cylinders equipped with poppet valves. They are inherently better balanced and have lighter reciprocating parts, hence are adapted for higher speeds. Because of their better balances they are applicable to conditions where foundations are not of the best. They take up less floor space, but obviously require more headroom than horizontal units.

Multi-cylinder engines can be arranged so that one group of cylinders exhausts at high back-pressure, supplying steam for process, while the other cylinders operate condensing or at some lower pressure, thus giving in effect a bleeder engine.

In deciding which of the engine types to select, one must consider, among other factors, space limitations, the equipment to be driven, use factor, steam temperature, whether or not exhaust will be used, steam economy, and cost. It is, of course, the object to select the most economical engine that conditions will permit, considering both fixed and operating charges.

Earlier in this article it was indicated that certain engine types cost more and are more efficient than others. Tables I and II show the approximate differences in cost and efficiency of the various types. Engine costs are for non-condensing engines designed for 125 to 175 lb. pressure and saturated steam. Costs do not include erection or generator.

In considering these cost data it should be remembered that the generator for a slow-speed engine will be somewhat more expensive than one for a high-speed unit.

Engine efficiencies given are the ratio of theoretical to actual steam rate for non-condensing engines operating with steam at pressures of 100 to 200 lb. They do not include mechanical efficiency or generator efficiency, which are listed separately. For condensing operation, the engine efficiencies given will be materially less (about 14% less for 26 in. vac.)

**Table I—Engine Costs in Dollars per Horsepower[1]**

| | Horsepower Rating | | | |
|---|---|---|---|---|
| | 100 | 200 | 500 | 1,000 |
| Single-Valve | 22 | 14 | 16 | .... |
| Four-valve, counterflow | 40 | 28.50 | 17.50 | 13.50 |
| Uniflow | 60 | 39.00 | 33.50 | 25.00 |

[1]Averages of prices from several manufacturers.

On page 308 is a chart from which the theoretical steam rate of an engine may be determined for any initial pressure or temperature and when exhausting at any back-pressure or vacuum. It is based on the heat energy available for work during adiabatic expansion of steam. This, of course, is not accomplished by the steam engine or any other present prime mover, hence actual steam rates are greater than those given by this chart. Approximate actual steam rates for different types of engine may be determined by dividing the theoretical steam rate found from the chart for the given initial and exhaust conditions by the product of the engine efficiency, mechanical efficiency and generator efficiency, assuming the engine is driving a generator. If the steam rate per horsepower-hour is required instead of per kilowatt-hour, divide the chart values for theoretical steam rate per horsepower by engine and mechanical efficiency.

If all of the engine exhaust will be used for heating or process during the entire year, there is no point in selecting an efficient and high-cost engine, for better dollar economy is obtained from a lower-cost, less-efficient engine. If, however, the ratio of process steam requirements to engine load is low it then becomes economical to pay more for an efficient engine. Likewise, if the engine is to operate at high load a large part of the time, a higher-cost engine is justified. Just how much more one can afford to pay should be deter-

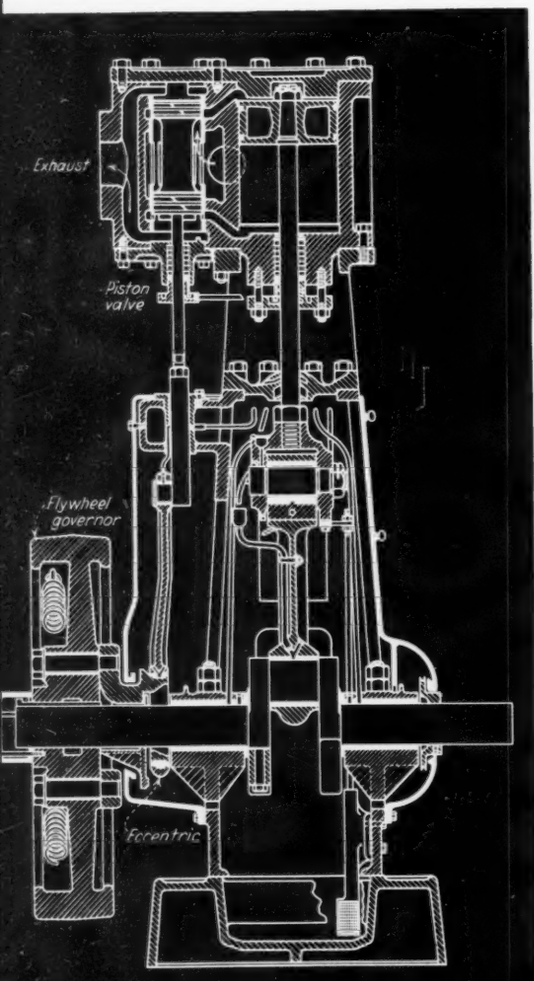



FIG.2

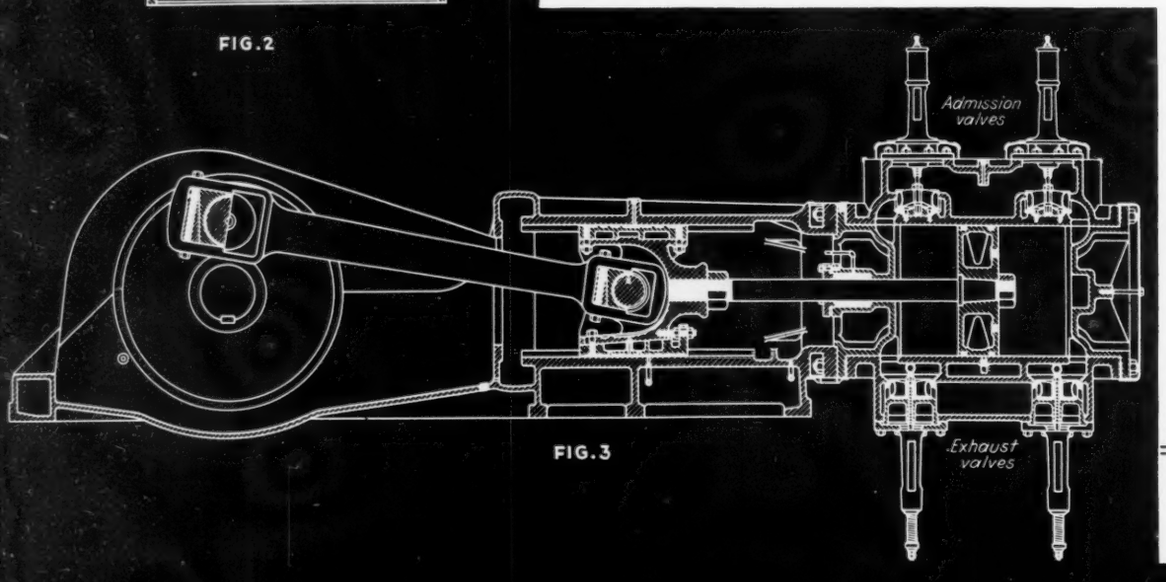



FIG.3

Fig. 2—Vertical slide-valve engine. Fig. 3—Horizontal 4-valve counterflow engine

Table II—Economy Data, Non-Condensing Engines[1]

| Hp. | Engine Efficiency on I. Hp., Per cent Single-Valve | Corliss | Uniflow | Mech. Eff. | Gen. Eff. |
|---|---|---|---|---|---|
| 100 | 51 | .... | .... | 90 | 93.2 |
| 200 | 56 | 65 | 73 | 90.5 | 93.4 |
| 300 | 59 | 68 | 76.5 | 91 | 93.5 |
| 400 | 61 | 70 | 79 | 91.5 | 94.0 |
| 600 | .. | 71.5 | 81.0 | 92.5 | 94.2 |
| 1,000 | .. | 73 | 83 | 93.5 | 94.5 |

[1]Averages from several manufacturers.

mined separately for each installation.

Superheat improves engine efficiency mainly because it decreases cylinder condensation. It increases efficiency about 1% for each 10 deg. of superheat. Uniflow engines are well adapted to take advantage of superheat. Superheat, however, is likely to interfere with the effectiveness of cylinder and valve lubrication. Consequently a high-grade oil that will not be decomposed by temperature should be used.

Change from saturated steam to superheat should be made with caution and not without consulting the engine builder. It may be that provision has not been made for the increased expansion that will take place with the higher temperatures; likewise the materials of which the cylinder and valves are constructed may not be appropriate.

If high superheat is used, an engine with poppet or piston valves is best. The two engines operating at 780 deg. have piston valves. At high superheat, slide valves tend to warp and are hard to lubricate. They should not be used with over about 100 to 150-deg. superheat.

Speed is an important characteristic of engine types. In general, engines are relatively slow-speed machines compared to turbines, which makes their application to d.c. generators and other slow-speed machines advantageous. Size of the unit, length of stroke, type of valve and valve gear are all factors in fixing maximum engine speed. Large engines with long strokes are always slow-speed machines. So also are units equipped with Corliss releasing gear. Poppet valves usually permit higher speeds than other types and vertical multi-cylinder engines operate as high as 325 r.p.m. for units of 1,000 hp. Smaller-capacity engines of this type operate at 400 to 600 r.p.m.

Engine speed is also limited by piston speed, which for short-stroke engines runs about 500 ft. per min., and for long-stroke engines about 800 ft. per min. Engines have, however, been built with piston speeds of nearly double these values.

One of the advantages of the steam engine is its high starting torque. Although this is of no particular significance when the engine is used to drive a generator it is of importance when a prime mover is considered for direct drive of hoists, mill machinery and reciprocating pumps.

Up to now, no American engine builder has developed an engine for high-pressure steam. It would seem that for modernization of industrial power plants there should be a field for a high-speed multi-cylinder vertical engine which would receive steam at say 600 to 1,000 lb. pressure and exhaust at high back-pressure to the existing prime mover. The engine inherently has good efficiency with high back-pressure, whereas the efficiency of the small turbine for high pressure decreases rapidly because of the small blade sizes required.

When purchasing an engine, one should, of course, first make sure that the purchase is being made from a reputable manufacturer. It will be wise to ask for a guaranteed steam-consumption data covering part loads as well as full load. Agreement should be reached before the purchase on how, where and when the acceptance tests are to be run and the manner in which the actual test steam rates are to be corrected in case initial and exhaust steam conditions are not exactly as contemplated in the guarantee data. It will also help to obtain an economical installation if the engine builder is given some idea of how high or low steam rates are to be evaluated. This will enable him to judge how much he can spend on details necessary for good efficiency.

Details of valve, gear and governors are given on pages 313 and 316.

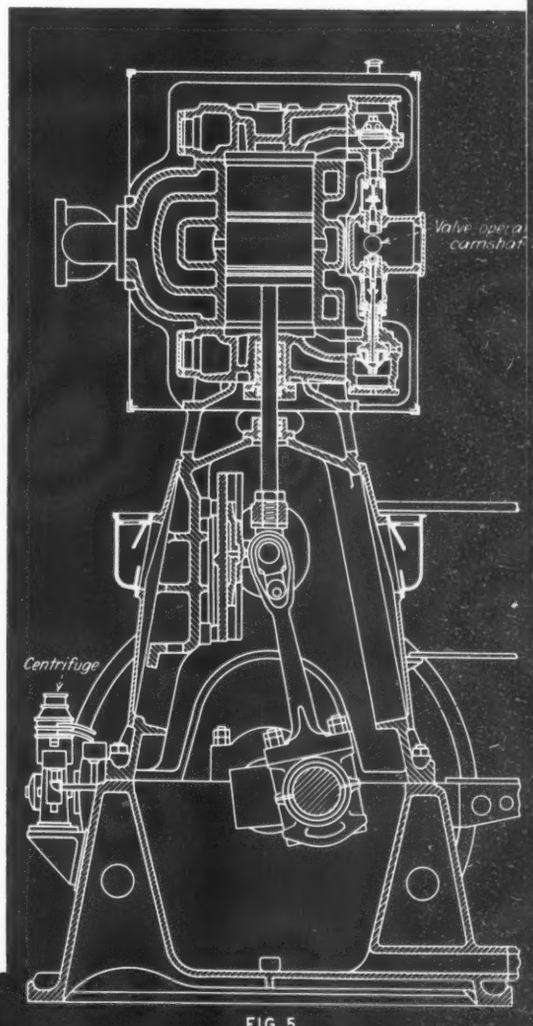



FIG. 5

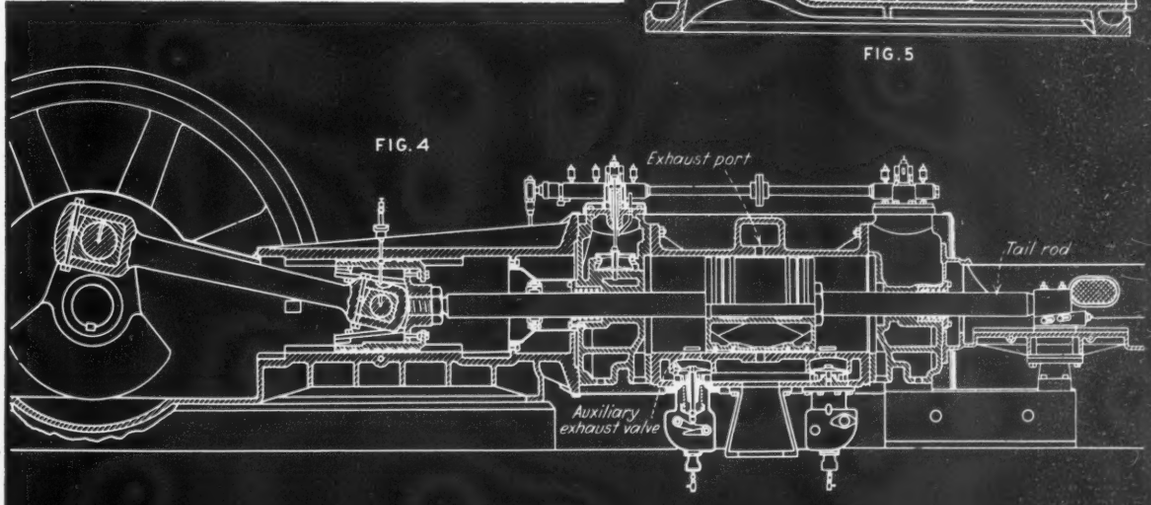



FIG. 4

Fig. 4—Horizontal uniflow non-condensing engine with auxiliary exhaust valves. Fig. 5—Multi-cylinder vertical condensing uniflow engine

# VALVES & GEAR

LAZY Humphrey Potter, valve tender on a mine pumping engine 200 years ago, found he could tie the steam-valve levers to the walking beam and the engine would run itself. That gave Humphrey a chance to sleep, and gave the world the first crude valve gear. Scores of valve-operating mechanisms and valves have been invented and discarded since then, but there are still many types in use. Some of the more common ones are here illustrated and described.

Small auxiliary steam engines still use single valves, such as the balanced slide valve of Fig. 5, operated by a single eccentric on the crankshaft, and the piston valve in Fig. 2 of the previous article. But with this valve governing is by throttling (an efficient method), and cutoff cannot be made to occur much earlier than about $\frac{5}{8}$ stroke. The riding cutoff, a second valve sliding on top of the main valve and operated by a second eccentric, gives separate control of cutoff, but is now seldom used. So prime movers must be provided with some other type of valve.

One of the early valve mechanisms, oscillating cylindrical valves and the releasing Corliss gear, has been modified for prime-mover governing. The arrangement of the four valves in the cylinder is shown in Fig. 9, and a modern design of Corliss release in Fig. 1. The cylindrical valves make it possible to decrease clearance volume below that necessary with slide valves; the four valves permit independent control and adjustment of engine events; the release mechanism allows governor control direct from a flyball governor; only one eccentric is required. One eccentric, however, forces cutoff to occur at half-stroke or earlier. For later cutoff, two eccentrics must be used. A rod from the eccentrics usually drives wristplates which operate the four valves.

In the Corliss gear of Fig. 1, rod *A* is oscillated by the eccentric to move arm *B* up and down. The latch on *B* engages valve arm *V*, which it lifts, thereby opening the admission valve. When trigger *T* engages cam *C*, the latch is disengaged and the valve closed by the dashpot to which *D* is connected. The position of *C* is controlled directly by the governor.

But the Corliss release mechanism cannot be operated at high speed. To get higher speeds, non-releasing toggle and cam operating gears have been developed for oscillating cylindrical valves like that in Fig. 4. Two eccentrics are required, one operating the two admission valves, the other the two exhaust valves. Governing is accomplished by varying the angular relation and throw of the eccentric driv-

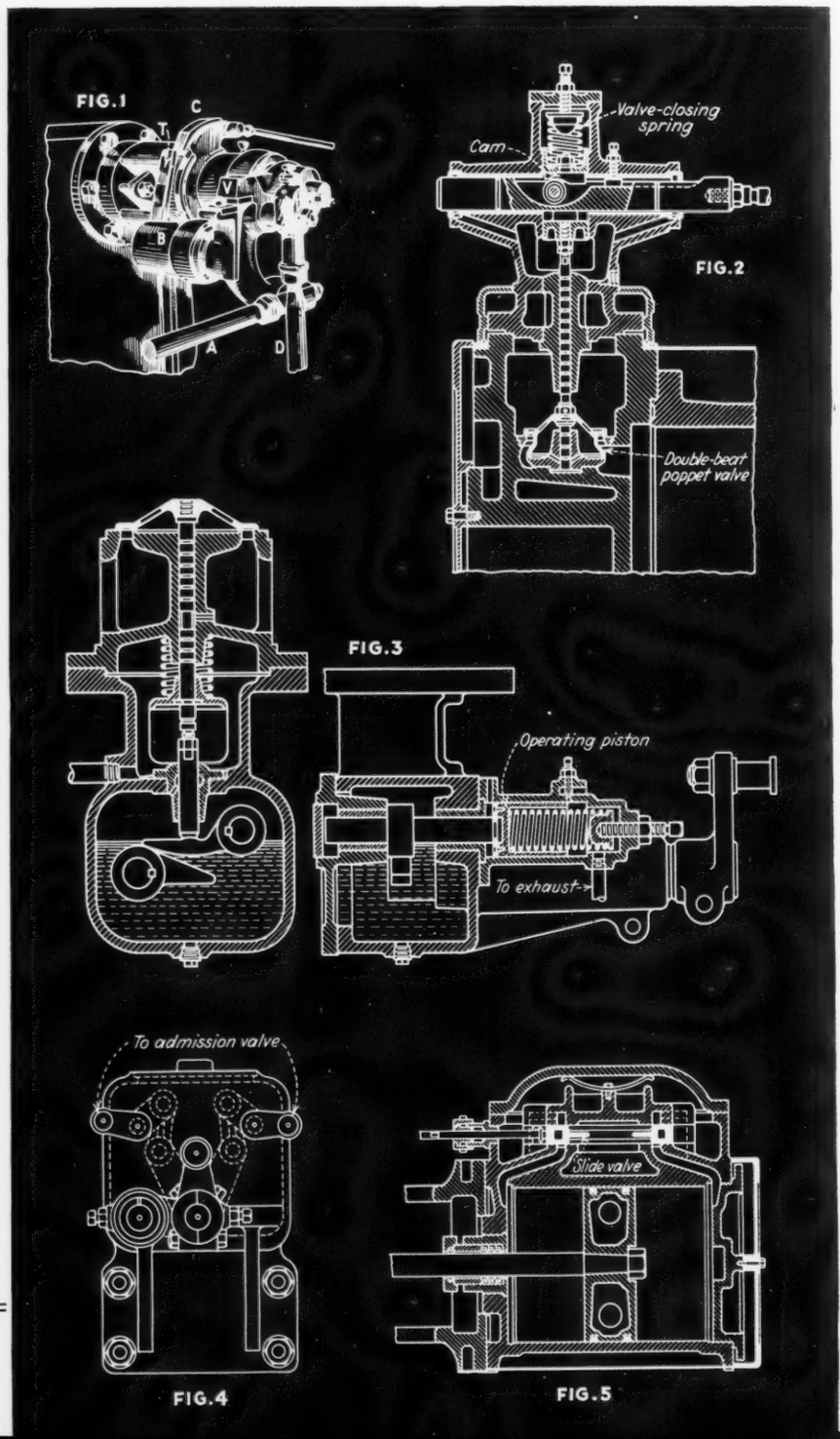



Fig. 1—Corliss release mechanism. Fig. 2—Poppet valve operated by reciprocating cam and spring. Fig. 3—Automatic auxiliary exhaust gear. Fig. 4—Toggle linkage for 4-valve engine. Fig. 5—Balanced slide valve

ing the admission valves, or by a linkage that does the same thing.

Many engine builders now use the poppet valve, which is adapted to higher speeds, has balanced valves which require less force to operate, and has a double seat to provide greater flow area and less wire-drawing. One type of poppet valve, made in two parts held together by a spring, is shown in Fig. 6. This arrangement permits steam tightness at both seats even though the cylinder metal expands at a different rate than the valve metal.

Poppet valves are operated by various rocker-arm and cam-lift mechanisms, as in Figs. 2, 7 and 8. In the design of Fig. 2, the ram rod, driven by the eccentric, carries a hardened roller which raises the cam and valve against a spring which closes the valve as the roller moves from under the cam. Long valve stems with labyrinth seals avoid stuffing boxes. The valve-operating cam of Fig. 8 is shaped so that the valve is both opened and closed by the cam, the spring being inserted between the balanced double-beat valve and the valve stem to give flexibility and assure alignment. Double eccentrics are required for cutoff governing as with non-releasing, oscillating valves.

A Corliss release mechanism can also be used to operate a drop-piston valve, as shown in Fig. 10. The piston uncovers two ports, so that it acts something like a double-beat poppet valve. This valve, spring-closed, avoids the problem of unequal expansion which may occur with poppet valves.

Valves of multi-cylinder vertical engines are usually driven by cams on horizontal shafts between the head- and crank-end valves. Fig. 4, page 313, shows a single camshaft operating the admission valves of a condensing uniflow engine. It is driven from the crankshaft by gears and an intermediate vertical shaft. Control of cutoff is obtained by shifting the camshaft axially, thereby causing valve-stem rollers to contact a different part of the cam. The valve gear of Fig. 7 is used on a multi-cylinder vertical engine with auxiliary exhaust valves for non-condensing operation. Shaft *A* is geared directly to the crankshaft and has a fixed angular relation to it. Cams controlling steam admission and opening and closing auxiliary exhaust valves are on shaft *A*. Cams for closing the admission valves, identical in shape to the opening cams, are on shaft *B*, driven by spiral gears from *A*. The angular relationship of *B* to *A* is controlled by the governor, which shifts *B* axially. If the governor advances the shaft, the roller following *B* drops before the roller following *A*, causing the valve to close and resulting in earlier cutoff.

Uniflow engines which are sometimes to operate non-condensing, sometimes condensing, may be provided with auxiliary exhaust valves that come into service automatically when vacuum is lost. Such an arrangement is sketched in Fig. 3. Vacuum acting on the small piston pulls the lifting cam out of operating position, allowing the auxiliary exhaust valve to remain closed while operating condensing. When the engine is running non-condensing, the spring forces the cam into operating position and the exhaust valve goes into action.

Common governor arrangements for controlling valve operation are described in the next article.

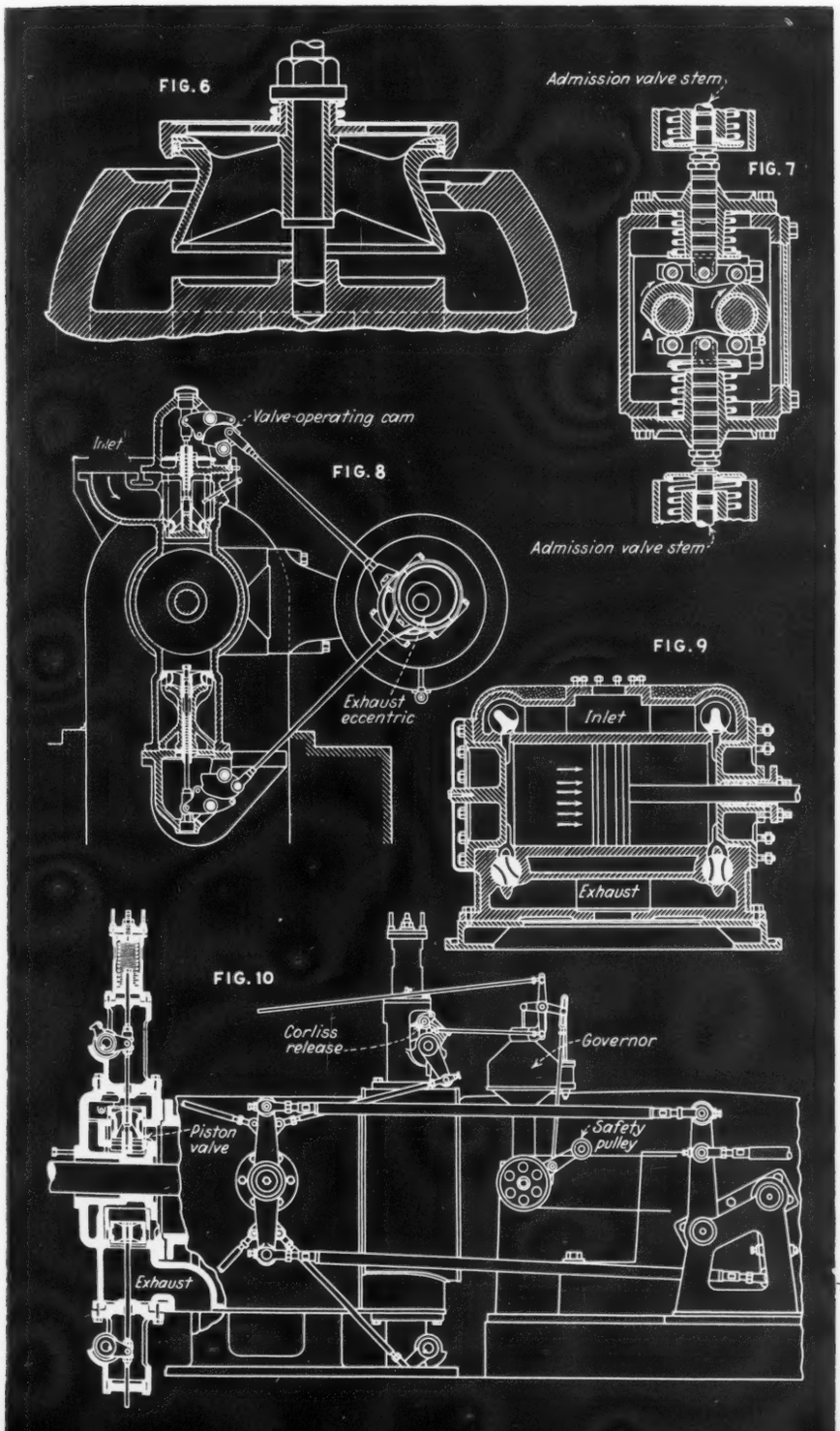



Fig. 6—Double-beat poppet valve with spring. Fig. 7—Variable cutoff gear for vertical engine. Fig. 8—Cam-operated poppet valves. Fig. 9—Arrangement of oscilalting cylindrical valves. Fig. 10—Drop piston valve and operating gear

# GOVERNING

STEAM engines may be governed either by throttling the steam supply before admission to the engine or by changing the point of cut-off. Throttle governing decreases the pressure of the steam supplied, and since the point of cut-off remains constant, the volume of steam used at each stroke remains the same at light load as at full load. In cut-off governing, admission pressure remains constant but volume of steam admitted to the cylinder is decreased as load decreases by causing cut-off to occur earlier in the stroke. Throttle governing is, therefore, not as economical as cutoff governing.

Methods used to change the point of cut-off depend upon the valve gear, and have been explained for several typical gears in "Valves & Valve Gear." In practically all types of engine gear, however, the governor acts directly; that is, oil or other types of relays are not used. It is important, therefore, that the governor be powerful enough to operate the linkages to which it is connected.

All speed governors on steam engines employ centrifugal force, and some a combination of centrifugal and inertia forces. Fig. 4 is an example of a flyball centrifugal governor on a multi-cylinder vertical engine. This governor is mounted directly on the valve camshaft and driven by gears from the engine shaft. At a certain speed the centrifugal force tending to throw the flyball out from the shaft is balanced by spring at the other end of the camshaft. An increase in speed tends to force the weight further out, moving the shaft axially and increasing compression of the spring until balance again obtains.

Figs. 2 and 3 illustrate flywheel inertia governors. Both of these operate to change the throw and angular relation of the eccentric that drives the admission valve gear. Centrifugal force acting on the governor weights is opposed in one case by a spiral spring, in the other case by a leaf spring operating on the principle previously explained. In addition, the inertia of the governor weight to any sudden change in speed acts to assist centrifugal force when speed increases, thus increasing the speed with which the governor reacts to sudden speed changes. When speed decreases, the inertia assists the spring, again aiding the governor action to increase speed.

When an engine-driven alternator is to operate in parallel with other units, some means must be provided to alter speed slightly while the unit is in operation so that the generator can be synchronized with the other units. Such speed changers are usually arranged to alter the tension in an auxiliary spring. Fig. 3 shows such an arrangement applied to a flywheel governor. Spring tension is changed by a small electric motor geared to a screw.

Fig. 1 shows an engine governed according to the pressure in the exhaust main, a provision which makes for economy when exhaust is used for process and other speed-governed engines can take electric load fluctuations.

Exhaust pressure is piped to the diaphragm *A*, which in turn operates the pilot valve of oil relay *B*. The piston of this relay raises or lowers the link block *C* to adjust cutoff according to pressure requirements. It is readily seen that when link block *C* is in line with the valve reach rod, movement of the rod is equal to the throw of the eccentric. When *C* moves downward, movement of the rod is less than the eccentric throw, and its movement with respect to the engine crank occurs earlier. This makes cutoff occur earlier, and the engine carries less load.

In addition to governors, most engines are provided with safety stops or overspeed stops. These provide automatic protection against overspeeding in case some part of the governor mechanism fails. In modern engines, they usually consist of a centrifugal device on the engine crankshaft arranged to trip the throttle valve at a predetermined engine speed.

Engines with belt-driven governors are arranged in various ways to shut the engine down in case the belt breaks or slips off the governor pulley.

Fig. 1—Engine governed by exhaust pressure. Fig. 2—Flywheel inertia governor with leaf spring. Fig. 3—Inertia governor with speed changer. Fig. 4—Flyball governor for vertical engine

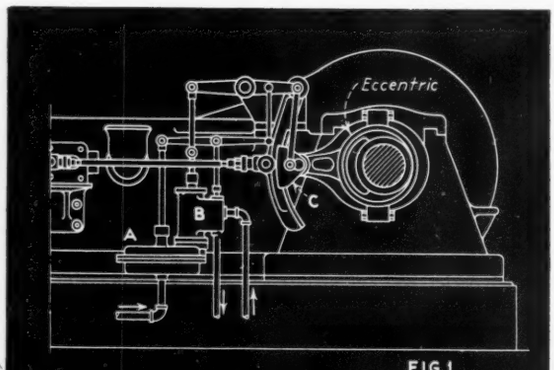


FIG.1

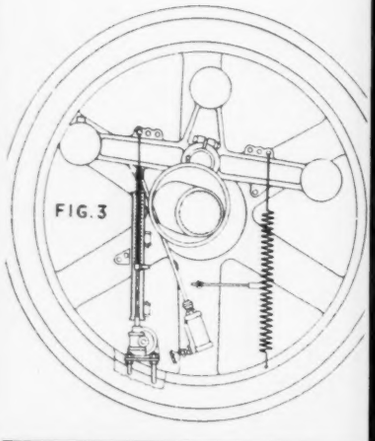


FIG.3

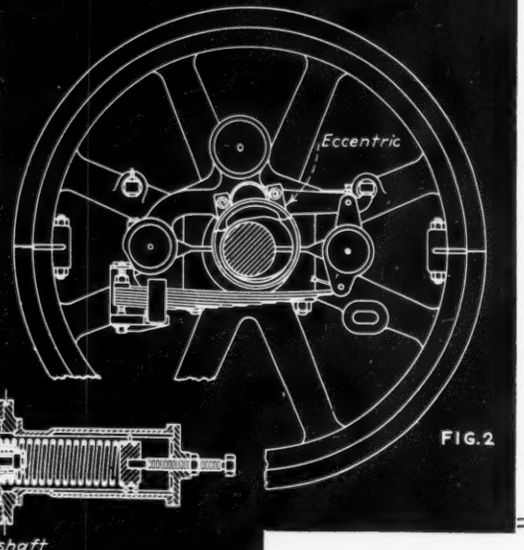


FIG.2

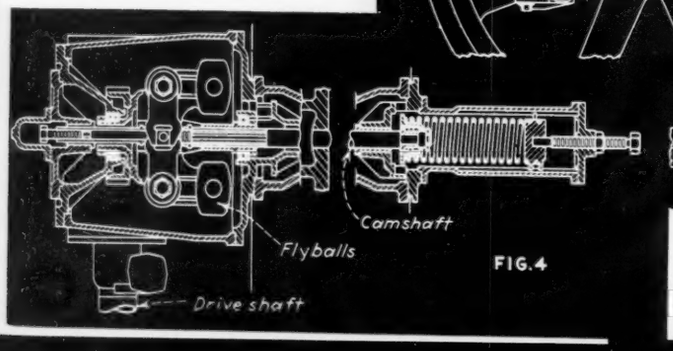


FIG.4

# DIESELS

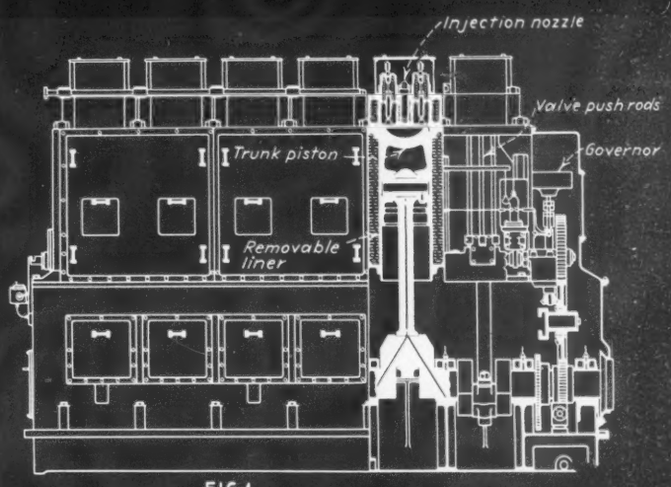



FIG.1

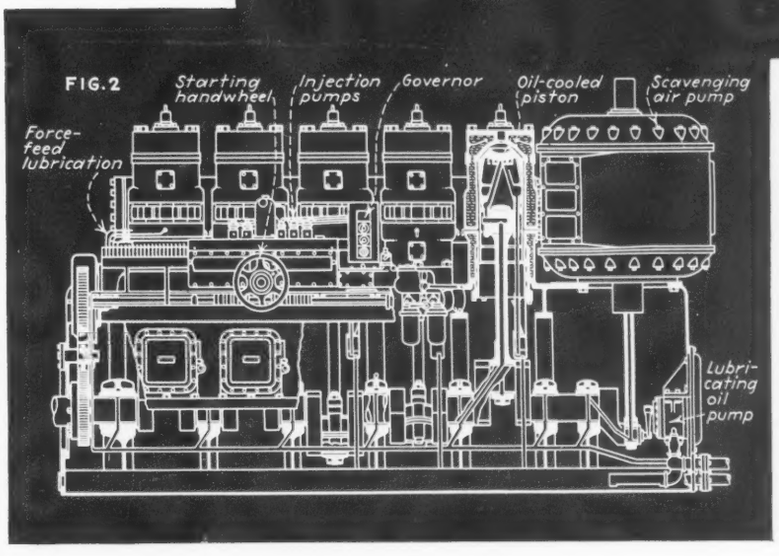



Fig. 1—Totally enclosed, 4-cycle unit. Fig. 2—2-cycle with oil-cooled piston

LET US assume that you have read "Looking Dieselward," June, 1935, *Power,* or have learned elsewhere the basic principles of the diesel, and are now principally interested in general features of engine types.

The modern diesel can be anything from about 3 hp. up to 22,500. One European builder says he can make them to 40,000 hp., given the orders. Both 2-stroke-cycle and 4-stroke-cycle engines are made, although most large engines are 2-cycle, most small ones 4-cycle. Above 25-in. bore, the 2-cycle shows decided advantages; below 8-in., the 4-cycle has shown best performance. Piston speeds vary from about 800-1,600 f.p.m. in small engines to 1,000-1,200 f.p.m. in large. Speeds in r.p.m. range from 2,000 for very small engines down to 125 or 150 for large, speeds (particularly in large engines to drive alternators) often being chosen for direct connection.

Air injection has been almost totally eliminated in large engines, except where previous prime movers in a a plant are of this type or where a particularly poor fuel is to be burned. Although its fuel may cost more, a solid-injection engine costs less to buy and run. Above 2,000 hp. and below 4,500, 2-cycle engines are favored almost 2 to 1, and above 4,500 hp., 2-cycle engines have the field to themselves. There are no 4-cycle, double-acting engines any more, but among large 2-cycle units, almost half are double-acting. Considering only capacities within the range of both types, it is clearly a choice between the simplicity and accessibility of the single-acting engine and the smaller space requirements and smaller number of cylinders for a given power of the double-acting. All large diesels are verticals.

In the intermediate range, whether 2-cycle or 4-cycle is a matter of choice. Many makers build both. The 2-cycle avoids valves, has smaller cylinders, takes less space for a given power, has lower first cost, is lighter, requires less maintenance. The 4-cycle is the standard design, is easier to cool and lubricate, requires no scavenging equipment, has lower m.e.p. Efficiencies are theoretically the same.

There are a number of special diesel designs, including crankless, opposed-piston, radial, and single-valved 4-cycle types. Two opposed-piston designs, one a crankless type, are sketched in Figs. 16 and 17. That in Fig. 16 uses a single crankshaft normal to the cylinder and just below it, piston thrust being transmitted to it through rockers. That in Fig. 17 has a straight shaft parallel to the cylinder, and piston thrust is transmitted through slippers to inclined disks (which are also flywheels) on each end. With this design, several cylinders can be grouped about the shaft and a generator driven between the two units. Piston crowns are flat and fuel is injected from one side. A radial type recently announced follows airplane-engine practice. There are also several V-type engines (to save crankcase cost and weight), and engines in which one or two power cylinders are replaced by ammonia compressor cylinders.

In this discussion, little or no reference is made to horizontal engines, because they are mainly used in oil-country applications, are usually low-powered and of the convertible type. Several are illustrated and described in "Gas Engines," pages 324-327.

In Europe, free-piston diesel-compressor sets have found acceptance, but thus far no American builder makes them. They have the usual opposed-piston diesel arrangement, but the outer end of each diesel piston forms a compressor piston; thus no crankshaft is required. The engines are started by racking the pistons apart.

In large diesels, fuel is sprayed by direct injection into the cylinder combustion space (smaller engines use various methods, Fig. 15). Fuel-pump plungers measure fuel delivered according to load at pressures of 4,000 to 8,000 lb. An auxiliary or "spill" valve is under control of the speed governor. To eliminate leaky packing, plungers now are hard-surfaced, working in lapped-fit, fine-grain, cast-

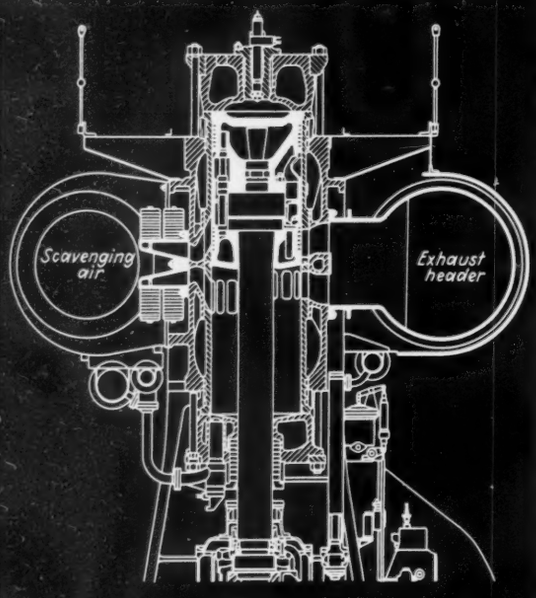



FIG. 3

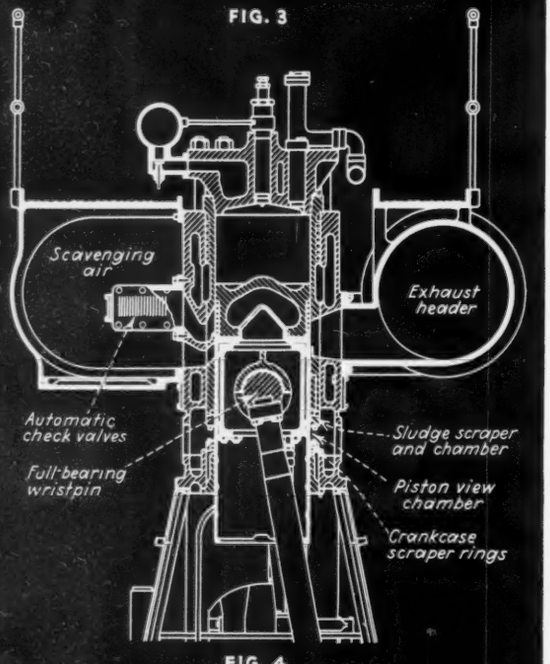



FIG. 4

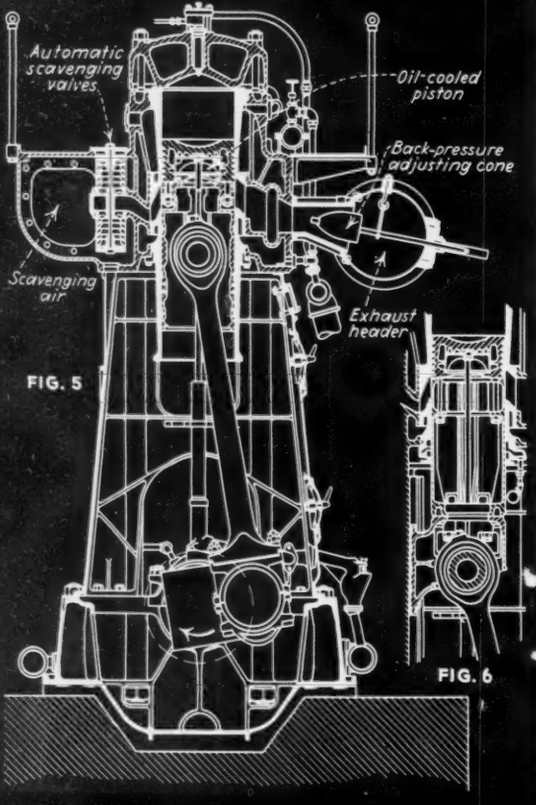



FIG. 5

FIG. 6

iron or hard-surface steel barrels. A separate pump—cam, plunger, barrel and valves—for each combustion space is almost universal. These may be near their respective cylinders, or grouped together, but in each case they alone measure the fuel and time in injection. If grouped, high-pressure piping is long and of various lengths, thus may introduce slight variations in injection timing. Fuel valves are not mechanically operated and timed, but are spring loaded, opening by fuel pressure and closing by spring. Valve lift is being cut to a minimum to insure sharp closing and thus eliminate "dribbling" of the last little bit of the fuel. If dribbled, fuel is not sprayed in, but hangs on the tip of the nozzle and is cracked there to form carbon. A fine strainer is put in the line at each fuel valve. Smaller diesels may use unit injection—combined fuel pumps and spray valves.

Fuel nozzles, like valves, are becoming more alike — multi-orifice (each 0.015 to 0.035 in. in diameter), with extremely careful machining to get drilled holes smooth, exactly spaced, rounded inside and sharp-edged outside. Proper cooling is essential. They must be quickly interchangeable. Machining facilities are now available to make exacting nozzles commercially.

Direct injection demands careful design of the combustion space. Two types are in use, the first impinging a short fuel jet on the piston top before atomization is complete, the impingement completing the work, the second using a long fuel jet carefully prevented from striking the piston. The first, by far most commonly used, utilizes a concave piston top. In the second, piston-head shape is dictated by the turbulence or non-turbulence desired.

There are almost as many ways of creating or avoiding turbulence as there are makers of engines, usually involving special shapes in the piston head and cylinder. A few examples are illustrated. That in Fig. 1 creates turbulence by causing air near the cylinder walls to slide down the concave piston-head surface and form an upward spout and whirl in the center. That in Fig. 2 causes air to be squeezed out between the tapered piston head and cylinder head, shooting up to the center of the cylinder head and whirling down into the concave crown. In each of these cases, fuel spray is almost horizontal. Combustion spaces on smaller engines vary even more, witness Fig. 15. The Hesselman-type engine, Fig. 20, injects fuel from one side toward a spark-plug at the other.

Scavenging through the cylinder head has been abandoned in large 2-cycle units. Port scavenging and exhaust through the cylinder walls is commonest, although some designs scavenge through ports and exhaust through valves in the head (the piston valve may act as an auxiliary power piston). Opposed-piston engines (about 1 unit in 12 among engines over 2,000 hp. is of this type) have scavenging ports at one end and exhaust ports at the other, following the uniflow principle. Each American builder uses a different system—more than 70 have been patented—to do this difficult job of cleaning and recharging the cylinders in the 1/10 sec. available (at 150 r.p.m.) while the ports are open. The design of Fig. 3 has scavenging ports uncovered before the exhaust port, a check valve preventing blowback until cylinder pressure drops below scavenging-air pressure. Then, on the upstroke, the exhaust port closes first so some supercharging can be obtained.

A reciprocating pump or rotary blower supplies scavenging air at $1\frac{1}{2}$ to $2\frac{1}{2}$ lb. to medium-size, 2-cycle engines. Large units may have a rotary blower attached to or driven by the engine, a centrifugal blower driven by electric motor, or a pump driven by an auxiliary diesel. Most units filter the air to reduce fouling of valves and pistons.

Cylinders are always fitted with removable nickel-chromium-cast-iron liners (Brinell 200-240) for these, with the pistons, are about the only major parts that wear out. Liner hardness, plus clean air and fuel, have cut wear in half during the last few years. Present wear occurs mainly at the combustion end where lubrication is poorest and gas pressure behind rings greatest. Chemical and physical properties of fuel oil (ash and water particularly) also play a part.

More medium-size and small diesels are using cylinder liners each

Fig. 3—Double-acting 2-cycle, 20,850-hp., 152-r.p.m. unit, second largest. Supercharging is obtained by using check valves. Fig. 4—Very large single-acting trunk-piston engine. Separate piston-pin housing gives full-length bearing and unpierced skirt. Annular chamber in lower cylinder shows failure of lubrication or cooling. Special long connecting rod. Fig. 5 — A 725-hp.-per-cyl. trunk-piston unit with exhaust-header cone to adjust backpressure, equalizing work and permitting some supercharging. Fig. 6—Crosshead design for the same unit, crosshead being on piston skirt to avoid a rod

year because of the difficulties involved in casting and machining the cheaper integral cylinder and jacket.

While pistons of small diesels are uncooled cast-iron or aluminum alloy, all large engines have cast-iron piston skirts and water or oil-cooled piston heads which may be cast-iron (Fig. 5), cast-steel, or forged-steel (Fig. 4). Rings are often of special materials or design, and include scraper or wiper rings to cut lube-oil consumption.

The tendency today is toward trunk-piston engines, eliminating the cross-head and reducing height about a third. This may make it necessary to provide oil scraper rings at the bottom of the liner in case the piston skirt comes below it at bottom dead center, or lubricating oil from the combustion chamber may be mixed with that in the sump. Many engines now use separate oiling systems for cylinders and bearings, centrifuging the cylinder oil to remove carbon before returning it to the sump, or keeping the oils separate. Cylinder oiling is usually by injection, timed to occur while the piston covers the oil outlet. There is a trend to lubricate nearer the combustion space.

Double-acting engines, of course, have a stuffing box through which the piston rod passes. Rods must be protected against combustion heat outside and possible cooling-medium corrosion inside. Shielding tubes of cast-iron are usually put outside to protect against combustion, and stainless steel tubes inside if water is the piston-cooling medium. If oil is used, no inside sleeve is required, but it may be necessary to provide wider bearings. In case part of the lubricating oil is used for cooling, about four times the amount necessary for lubrication is delivered.

With oil cooling of pistons, oil should be circulated for a while after

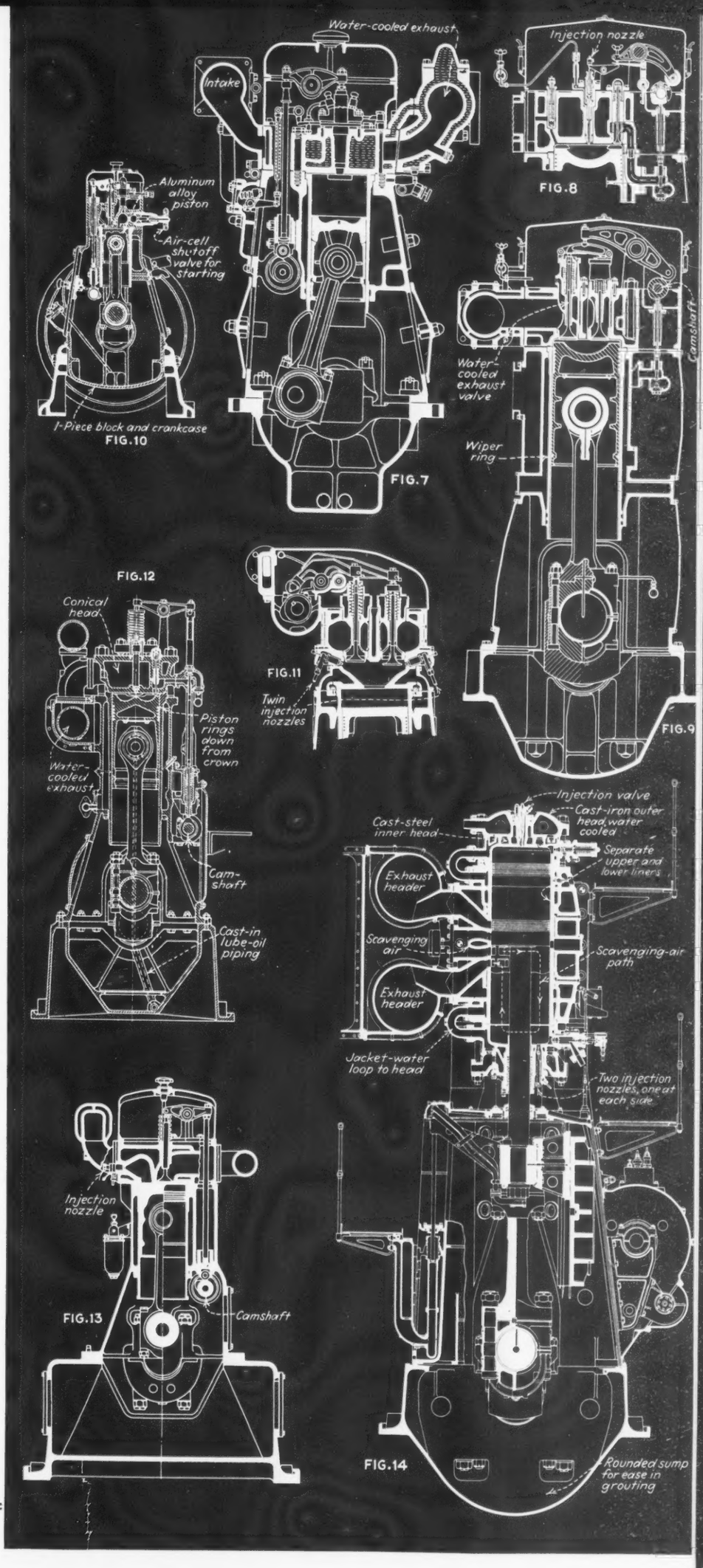



Fig. 7 — Compact, light-weight, 4-cycle, 1,400-r.p.m. unit also made as 12- and 16-cyl. V-type. A 5½x7-in. cyl. gives 25 hp. at this speed. Fig. 8—Detail of engine in Fig. 9, showing cross-section through nozzle. Fig. 9—A 4-cycle, 400-r.p.m., 12½x15-in. unit with dual valves to reduce size, water-cooled valve cages, wedge-action governor varying valve lift. Fig. 10—A little diesel, going as low as 3 hp. Vertical injection into a circular combustion chamber with side air cell. Flat-topped piston almost touches cylinder head. Hand started. Fig. 11—Duplex-injection 4-cycle unit. For frame details, see page 326. Fig. 12—4-cycle trunk-piston with conical nickel-iron cylinder head seating on liner without gaskets. Fig. 13—4-cycle, 7x8-in., 270-hp. 1,200-r.p.m. unit. Side fuel injection into a pear-shaped chamber. Fig. 14—Largest American diesel, (7,000-hp.). Exhaust ports between scavenging ports and water-cooled cylinder head

the engine is shut down, otherwise residual heat in the head at the time of shut-down may form carbon just under the piston crown. Some designs avoid this by spraying the oil into the piston and draining it through a base hole.

The engine frame itself consists of bedplate, frames and cylinder block, usually held together by tie rods, (some have heavier construction without the ties). While in small engines heads may be enbloc, on large units they are separate.

Increasingly, auxiliaries such as oil and air filters, fuel-transfer pumps, lubricating-oil pumps and cooling-water pump, are installed within the engine inclosure.

Supercharging (supplying intake air under pressure) is advantageous to increase power output per cylinder. Two-cycle design makes it difficult to supercharge economically, so only a few 2-cycle units are supercharged. Usually they use the Buechi system—a centrifugal blower driven by engine exhaust gases. Thus far, only a few American diesels have been supercharged, and some authorities feel that the future field for supercharging is in small and medium 4-cycle engines running at medium and high speeds.

In stationary engines, weight makes little difference, so cast-iron is still the commonest material. For motive-power or marine service, however, weight is important. Engines for both uses often have aluminum alloy pistons and welded steel plate and casting or aluminum-alloy cast frames. The welded frame is less rigid and more likely to be noisy, but silencing manifolds minimize the latter, proper design the former. Aluminum alloy halves the weight but more than doubles the cost. A 2-cycle, V-type engine uses a frame composed of two deck-plates and side and end plates welded to main crossframes. Bored holes in the deck plates receive the double-walled cast liners, which incorporate the cooling passages.

Many engines are direct-connected to generator or compressor, hence run at almost constant speed, although load may be highly variable. This stresses division of load equally between cylinders and proper governing. Where there is speed variation, the minimum will usually be 30 to 40% of rated speed, and a heavier flywheel is provided.

Governing was once a problem, but increasing application of diesels to a.c. generation in municipal and industrial plants concentrated attention on this. The result is that governing can be as close as it is with any other prime mover. "Governing," page 329, takes up the principles, and page 331 shows four governor types from the simple flyball to the complicated and accurate isochronous type which can control a whole plant. All control the quantity of fuel injected and are driven from the crankshaft, a layshaft or the fuel-pump driveshaft. The throttle is set to feed enough fuel to give the desired feed and the governor controls an auxiliary or spill valve, or rotates the fuel-pump plunger to vary quantity, timing or pressure.

Safety devices and instruments often fitted, include pressure and temperature gages and controls on lubricating oil, cooling water (except on units) and exhaust, plus flow indicators on the first two. Pyrometers also show engine performance. Automatic plants use controls that stop the engine if any condition reaches a danger zone, signalling the operator that the engine is down. Such devices include high-temperature or pressure elements on lubricating oil, jacket water, exhaust and bearings; low-water alarms; and under- or overspeed stops.

Automatic plants are being installed here and there—for standby or emergency service or to avoid the cost of a full-time attendant. Because of high American wage rates, a full-time attendant's salary may seriously affect operating costs of a small diesel plant, so a semi-automatic plant may be cheaper.

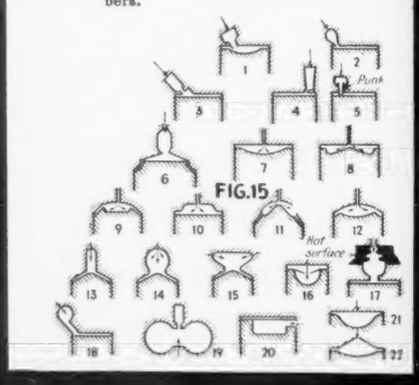
Stationary plant cooling systems are increasingly of the closed type, particularly since more favorable insurance rates have been inaugurated on this type of plant. Heat may be recovered by heat exchangers.

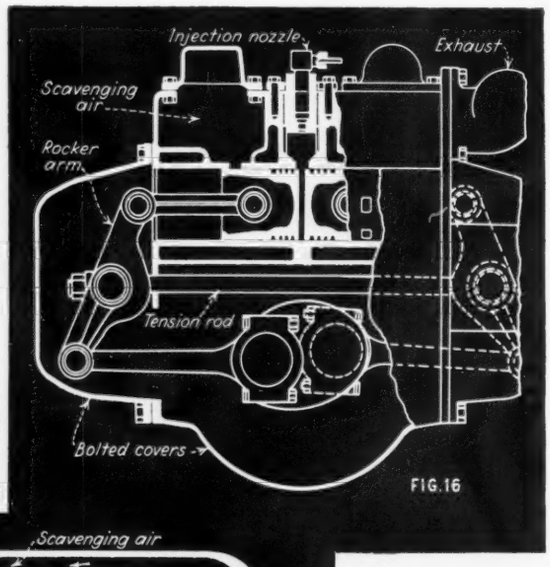
Diesels in cities, etc., must be silenced and isolated to prevent noise and vibration. Adequate intake and exhaust silencers are available, as well as vibration-isolating bases utilizing cork, helical springs, etc.

While diesel fuel consumption is fairly constant over a wide range of size and load, about 0.4 lb. per b.hp.-hr., small engines may increase this to 0.45 and large ones decrease it to 0.35. In fact, one foreign 5,500-hp. engine gets 0.329 with 18,000-B.t.u. fuel, an efficiency of 41%. Driving generators, large engines will deliver 13½ to 14 net kw.-hr. per gal. of fuel at full load, smaller engines about 12.

Heat in the fuel goes about 35% to power, 30% to jacket water and 35% to exhaust and radiation. As much as 85% of the total heat may be used. For methods, see the *Power* series on waste heat, June, July, Aug. and Sept., 1933. Fuels are discussed in "I.C. Fuels," page 299.

Fig. 15—More combustion-chamber shapes. 1, 2, 3, 4, 5, 6—precombustion chambers, 7—4-cycle quiescent, 8—4-cycle turbulent, 9, 10—2-cycle quiescent, 11, 12—2-cycle semi-turbulent, 13, 14—single- and multi-orifice turbulent, 15—twin-spray turbulent, 16—turbulent with division plate, 17—semidiesel with center hot surface, 18—separate rotary turbulent, 19—double-pocket with air cell, 20—3-dimensional turbulent, 21, 22—air-injection chambers.



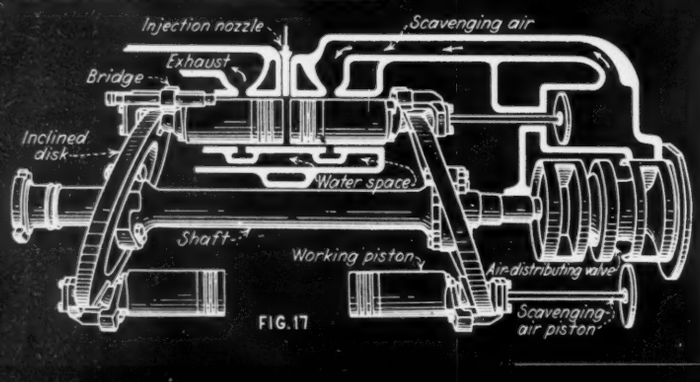




Fig. 16 and 17—Two opposed-piston designs. Fig. 16 has a rocker-arm arrangement. Cylinder, jacket, scavenging chamber, exhaust passage, crankcase and combustion chamber are cast integral. Fig. 17 is a crankless opposed-piston unit utilizing welding

# DIESEL UNITS

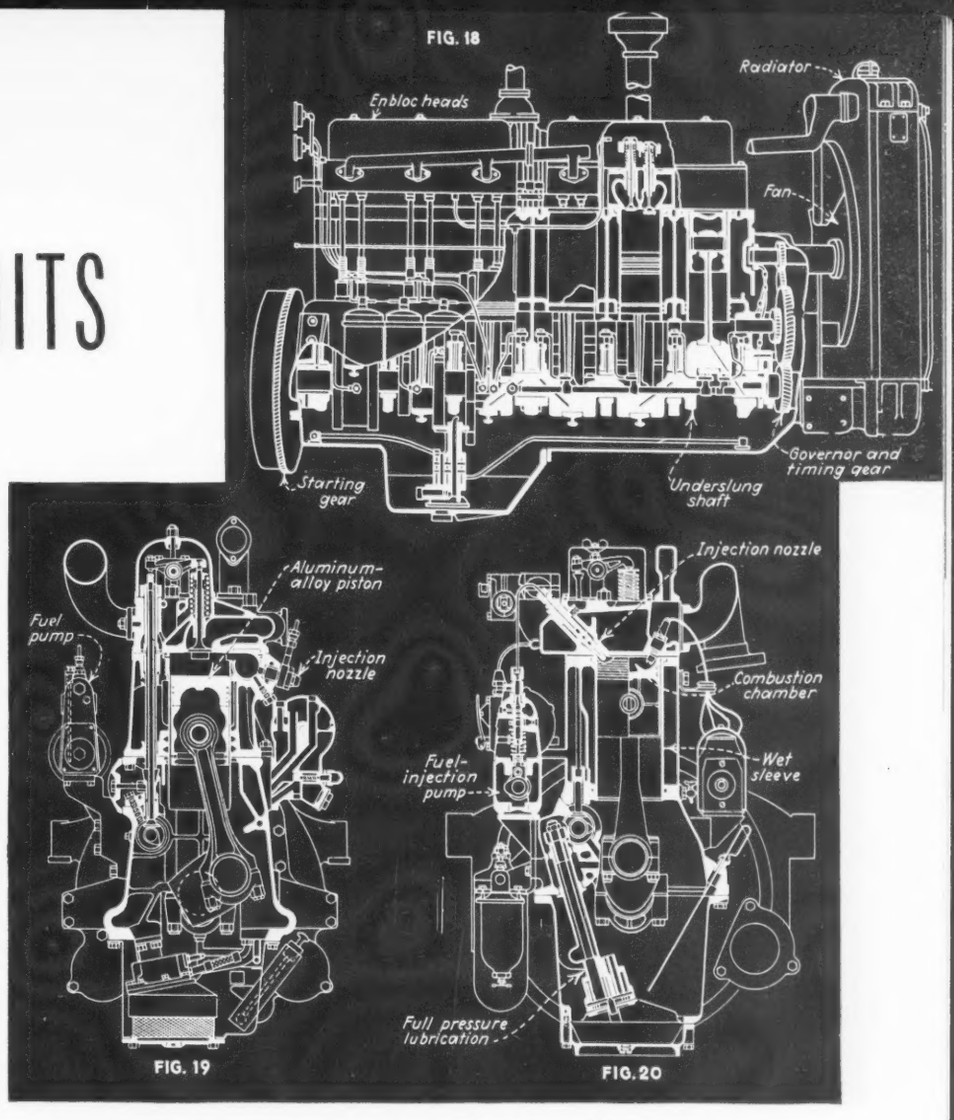



Fig. 18—Unit with separate liners, integral radiator, enbloc heads, etc. Fig. 19—Side-injection unit. Fig. 20—Hesselman unit (not a diesel)

DURING the last five years, there has been a phenomenal rise in the application of the diesel "power unit," meaning a small diesel engine complete with oil and air filters, starting element, cooling system (including fan and radiator), controls, instrument panel, and a base or sub-base to carry the engine and a direct-connected generator. An outgrowth of marine and mobile-machinery demands, these units provide power for small plants or buildings; auxiliary drives; standby or emergency sets; temporary and semi-portable plants; peak, lighting or refrigeration units, etc.

These units are usually 4-cycle verticals, although there are a few opposed-piston and V-type sets. Many go to more cylinders of smaller bore for better balance and smooth operation over a wide range of speeds; others use heavier flywheels. Four or six cylinders are preferred numbers. Starting may be by air, electric storage battery, auxiliary gasoline engine or hand cranking. Some are converted to gasoline for the first few revolutions. Hand cranking is limited to engines of about 40 hp. and less, and air is seldom used in powers that low. Storage batteries are of particular advantage where the set is installed to operate automatically.

Combustion systems in almost endless variety are used (see Fig. 15): turbulent air motion in the cylinder with an auxiliary air cell, semi-turbulent, separate rotary combustion system, direct injection (either vertically at center or side, at an angle, or from the side), precombustion chamber, or Hesselman or semi-diesel operation. Hesselman types use battery or high-tension magneto with impulse coupling and have built-in governors controlling a throttle valve in the air-intake manifold. If individual pump injection is used, pumps are grouped for ease in driving. Some designs combine fuel pump and valve, thus eliminating high-pressure piping.

Cast-iron is still the commonest frame material, although some makers have gone to aluminum alloy (10% less weight) or welded frames. The engines are heavier than gasoline engines of the same size, but "hang on" to a load better at reduced speeds. R.p.m. and piston speeds are comparable—piston speeds may reach 1,900 f.p.m., r.p.m. may reach 2,400. In five years, speed has been increased about 60%, piston speed about 30, while m.e.p. is up 10%.

Most of these engines follow automotive practice, using overhead valves, enbloc construction (although block and crankcase may be separate), underslung crankshaft, and crankshaft in the crankcase operating valves through pushrods. The engine may or may not have liners and valve-seat inserts. The flywheel is usually enclosed. Pistons may be either cast-iron or aluminum alloy, usually with a heavy crown (sometimes a heat-resisting insert) and ring belt and a ribbed head for better temperature gradient. The top ring is down from the piston top to reduce wear and prevent sticking. Rings, although they carry a much greater gas load than gasoline-engine rings, are the same width. Governors are built in, usually at the front end and driven from the crankshaft, and control the amount of fuel injected.

There are also several special engine shapes which come in this group, among them the V type and the opposed-piston units. V-type construction provides a shorter, wider engine for a given power and reduces crankcase size, weight and cost as well as injection-piping distances. It is common to put the fuel pumps inside the V, as well as the oil filters and other auxiliaries.

Power takeoff through a clutch on the flywheel (enabling the engine to be started unloaded) or through stub-shafts is used, as well as direct connection to a generator, pump, blower or compressor.

# 190 TYPICAL NEW DIESEL PLANTS

## 307 Engines Installed Since April 1, 1935

| Owner | Location | Service | Equipt. Driven | Engine Builder | No. Units | Horse-power Each | Speed, R.P.M. | No. Cyl. | Cyl. Bore | Cyl. Stroke | Cycle | Starting Method | Fuel Injection | Fuel Consump. lb. per h. hp. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| **1,000 Hp. or More per Engine** | | | | | | | | | | | | | | |
| Freeport | N. Y. | Mun. Add. | Alt. | BuS | 1 | 3,000 | 240 | 10 | 19½ | 27 | 2 | A | M | 0.40 |
| Rockville Centre | N. Y. | Mun. Pwr. | Alt. | MeI | 1 | 2,865 | 120 | 8 | 29 | 48 | 4 | A | A | 0.40 |
| Hilo Elec. Co. | Hilo, T. H. | C. S. Add. | Alt. | BuS | 1 | 2,400 | 240 | 8 | 19¾ | 27 | 2 | A | M | 0.39 |
| Carthage | Mo. | Mun. Add. | Alt. | Nor | 1 | 2,250 | 225 | 6 | 21 | 29 | 2 | A | M | .... |
| Ill. Cent. R.R. | Chicago, Ill. | Locomotive | Gen. | BuS | 1 | 2,000 | 550 | 10 | 14 | 16 | 2 | E | M | 0.41 |
| Logan City | Utah | Mun. Pwr. | Alt. | BuS | 1 | 1,800 | 240 | 6 | 19½ | 27 | 2 | A | M | 0.39 |
| | | | Pump | BuS | 1 | 1,700 | 250 | 6 | 19½ | 27 | 2 | A | M | 0.40 |
| U. S. Dist. Eng. | Louisville, Ky. | Dredge | Gen. | ..... | 1 | 840 | 250 | 6 | 15⅝ | 21 | 2 | A | M | 0.40 |
| | | | Prop. | ..... | 2 | 750 | 250 | 6 | 15⅝ | 21 | 2 | A | M | 0.40 |
| Lubbock | Tex. | Mun. Pwr. | Alt. | Nor | 1 | 1,665 | 225 | 8 | 17 | 25 | 2 | A | A | .... |
| Russel | Kans. | Lt. & Pwr. | Gen. | DLV | 1 | 1,600 | 200 | 8 | 22 | 30 | 4 | A. | M | 0.41 |
| Sioux Falls | S. D. | Mun. Pwr. | Alt. | Nor | 1 | 1,500 | 257 | 4 | 17 | 25 | 2 | A | M | .... |
| U. S. Dept. of Int. | Coolidge, Ariz. | Standby | Alt. | Nor | 2 | 1,400 | 257 | 7 | 16¾ | 23 | 2 | A | M | .... |
| Longmont | Colo. | Mun. Add. | Gen. | FM | 1 | 1,400 | 300 | 8 | 16 | 20 | 2 | A | M | 0.39 |
| Marshall | Mich. | Mun. Pwr. | Alt. | Nor | 1 | 1,250 | 257 | 5 | 17 | 25 | 4 | A | A | .... |
| Sikeston | Mo. | Mun. Pwr. | Alt. | Nor | 1 | 1,250 | 257 | 5 | 17 | 25 | 2 | A | M | .... |
| Lafayette | La. | Mun. Add. | Gen. | FM | 1 | 1,225 | 300 | 7 | 16 | 20 | 2 | A | M | 0.39 |
| Vero Beach | Fla. | Mun. Pwr. | Alt. | MeI | 1 | 1,090 | 225 | 8 | 17½ | 25 | 4 | A | M | 0.36 |
| Denton | Tex. | Lt. & Pwr. | Gen. | DLV | 1 | 1,000 | 240 | 8 | 17½ | 24 | 4 | A | M | 0.41 |
| Easton | Md. | Mun. Pwr. | Alt. | BuS | 1 | 1,000 | 300 | 8 | 16¼ | 21 | 4 | A | M | 0.39 |
| Harlan | Iowa | Mun. Add. | Gen. | Ful | 1 | 1,000 | 225 | 8 | 17½ | 24½ | 4 | A | A | .... |
| Madison | S. D. | Mun. Pwr. | Alt. | BuS | 1 | 1,000 | 300 | 8 | 16¼ | 21 | 4 | A | M | 0.39 |
| Thief River Falls | Minn. | Mun. Pwr. | Alt. | BuS | 1 | 1,000 | 300 | 8 | 16¼ | 21 | 4 | A | M | 0.39 |
| **500-875 Hp. per Engine** | | | | | | | | | | | | | | |
| Sturgis | Mich. | Mun. Pwr. | Alt. | BuS | 2 | 875 | 300 | 5 | 15⅝ | 21 | 4 | A | M | 0.40 |
| Cushing | Okla. | Mun. Pwr. | Alt. | MeI | 3 | 750 | 225 | 6 | 17½ | 25 | 4 | A | M | 0.35 |
| Potash Co. of Amer. | Carlsbad, N. M. | Pwr. | Gen. | Nor | 1 | 750 | 257 | 3 | 17 | 24 | 2 | A | M | .... |
| | | | | DLV | 1 | 750 | 180 | 4 | 22 | 30 | 4 | A | M | 0.43 |
| Parke-Davis & Co. | Detroit, Mich. | Pwr. | Gen. | BuS | 1 | 750 | 300 | 6 | 16¼ | 21 | 4 | A | M | 0.38 |
| Hotel New Yorker | New York, N. Y. | Pwr. Add. | Gen. | BuS | 1 | 750 | 300 | 6 | 16¼ | 21 | 4 | A | M | 0.39 |
| Vinton | Iowa | Mun. Add. | Gen. | FM | 1 | 750 | 257 | 5 | 16 | 20 | 2 | A | M | 0.39 |
| Eastport | Maine | Pwr. | Alt. | Win | 2 | 700 | 360 | 8 | 14 | 16 | 4 | A | M | 0.39 |
| St. Louis | Mich. | Mun. Add. | Gen. | FM | 1 | 700 | 300 | 4 | 16 | 20 | 2 | A | M | 0.39 |
| N. Y., N. H. & Htfd. R.R. | New Haven, Conn. | Loco. | Gen. | CB | 5 | 660 | 750 | 8 | 10½ | 12 | 4 | A | M | 0.40 |
| Baldwin Loco. Works | Eddystone, Pa. | Pwr. | Loco. | DLV | 1 | 660 | 600 | 6 | 12½ | 15½ | 4 | A | M | 0.42 |
| Inter. Mining Co. | Mexico | Pwr. | Gen. | DLV | 1 | 660 | 240 | 6 | 17 | 24 | 4 | A | M | 0.43 |
| Bowie | Tex. | Mun. Add. | Gen. | FM | 1 | 630 | 300 | 6 | 14 | 17 | 2 | A | M | 0.40 |
| Hutchinson | Minn. | Mun. Pwr. | Alt. | MeI | 2 | 625 | 225 | 5 | 17½ | 25 | 4 | A | M | 0.36 |
| Blair | Neb. | Mun. Pwr. | Alt. | BuS | 1 | 625 | 300 | 5 | 16¼ | 21 | 4 | A | M | 0.39 |
| Lake Mills | Minn. | Mun. Add. | Gen. | FM | 1 | 600 | 400 | 6 | 12 | 15 | 2 | A | M | 0.40 |
| Fla. Pub. Util. Co. | Marianna, Fla. | C. S. | Alt. | BuS | 1 | 600 | 225 | 6 | 16¼ | 23 | 4 | A | A | 0.42 |
| W. M. Hooper & Sons | Baltimore, Md. | Pwr. Add. | Alt. | MeI | 1 | 600 | 214 | 5 | 17½ | 25 | 4 | A | M | 0.36 |
| Key West Elec. Co. | Key West, Fla. | C. S. Add. | Alt. | Nor | 1 | 600 | 240 | 3 | 16¼ | 23 | 2 | A | A | .... |
| Minter City Oil Mill | Miss. | Pwr. | ..... | FM | 1 | 600 | 400 | 6 | 12 | 15 | 2 | A | M | 0.40 |
| Pato Consol. Gold Dredging Co. | San Francisco, Calif. | Dredge | Pump | FM | 1 | 600 | 400 | 6 | 12 | 15 | 2 | A | M | 0.40 |
| Hammon Consol. Gold Fields | Nome, Alaska | Pwr. | Gen. | Pac. | 6 | 525 | 225 | 6 | 15 | 24 | 4 | A | A | .... |
| | | | | FM | 2 | 525 | 300 | 5 | 14 | 17 | 4 | A | A | .... |
| | | | | | 4 | 525 | 300 | 5 | 14 | 17 | 2 | A | M | 0.40 |
| East Side Levee & Sanit. Dist. | E. St. Louis, Ill. | Drainage | Pump | FM | 1 | 225 | 400 | 6 | 10 | 12½ | 2 | A | M | 0.40 |
| | | | | | 1 | 100 | 400 | 4 | 8¼ | 10½ | 2 | A | M | 0.42 |
| Benson | Minn. | Mun. Add. | Gen. | FM | 1 | 500 | 400 | 5 | 12 | 15 | 2 | A | M | 0.40 |
| **200-480 Hp. per Engine** | | | | | | | | | | | | | | |
| Shelbina | Mo. | Lt. & Pwr. | Gen. | DLV | 1 | 480 | 300 | 6 | 15 | 18½ | 4 | A | M | 0.40 |
| Salisbury | Mo. | Mun. Pwr. | Alt. | MeI | 1 | 460 | 300 | 6 | 12½ | 22 | 4 | A | M | 0.38 |
| Green-Mtn.-Adirdk. Ferry Co. | Plattsburg, N. Y. | Ferry | Gen. | CB | 2 | 450 | 360 | 8 | 11½ | 15 | 4 | A | M | 0.40 |
| Elko-Lamoille Power Co. | Elko, Nev. | C. S. | Alt. | BuS | 1 | 450 | 360 | 6 | 13 | 17 | 4 | A | A | 0.43 |
| Imperial | Neb. | Mun. Pwr. | Alt. | FM | 1 | 420 | 277 | 6 | 14 | 17 | 2 | A | M | 0.40 |
| Wray | Colo. | Mun. Add. | Gen. | FM | 1 | 420 | 300 | 6 | 14 | 17 | 2 | A | M | 0.38 |
| Orange County Water Works | Orange, Tex. | Irrig. | Pump | MeI | 4 | 400 | 315 | 5 | 12½ | 22 | 4 | A | M | 0.39 |
| Burlington | Kans. | Lt. & Pwr. | Gen. | FM | 2 | 375 | 300 | 5 | 14 | 17 | 2 | A | M | 0.38 |
| Windom | Minn. | Mun. Pwr. | Alt. | BuS | 2 | 375 | 360 | 5 | 13 | 17 | 4 | A | M | 0.40 |
| Kimball | Neb. | Mun. Pwr. | Alt. | MeI | 1 | 375 | 225 | 3 | 17½ | 25 | 4 | A | A | 0.40 |
| Connersville | Ind. | Mun. Pwr. | Gen. | CB | 2 | 363 | 360 | 6 | 11½ | 15 | 4 | A | M | 0.40 |
| Deshler | Neb. | Mun. Pwr. | Alt. | FM | 1 | 350 | 360 | 5 | 14 | 17 | 2 | A | M | 0.38 |
| | | | | | 1 | 165 | 300 | 3 | 12 | 14 | 2 | A | M | 0.38 |
| Miller | S. D. | Pwr. | Alt. | FM | 1 | 350 | 300 | 5 | 14 | 17 | 2 | A | M | 0.38 |
| Wrangel | Alaska | Lt. & Pwr. | Gen. | FM | 1 | 350 | 300 | 5 | 14 | 17 | 2 | A | M | 0.38 |
| Witt Ice & Gas Co. | Los Angeles, Cal. | Plt. Pwr. | Gen. | CB | 1 | 325 | 350 | 8 | 10½ | 12 | 4 | A | M | 0.40 |
| Anglo-Mex. Petrol. Co. | Aguilla, Tamp., Mex. | Plt. Pwr. | Gen. | CB | 6 | 315 | 315 | 6 | 11½ | 15 | 4 | A | M | 0.40 |
| The Namm Store | Brooklyn, N. Y. | Pwr. Lt., | Alt. & Comp. | Wor | 4 | 300 | 514 | 6 | 10¾ | 14½ | 4 | A | M | 0.39 |
| | | Air Cond. | Alt. | Wor | 1 | 150 | 514 | 6 | 8 | 10½ | 4 | A | M | 0.39 |
| Defiance Screw Mach. Prod. Co. | Defiance, Ohio | Pwr. | Gen. | Buc | 2 | 300 | 360 | 4 | 9½ | 14 | 4 | A | M | 0.40 |
| Fate-Root-Heath Co. | Plymouth, Ohio | Loco. | Gen. | CB | 1 | 300 | 650 | 6 | 8 | 13½ | 4 | A | M | 0.40 |
| La Porte City | Iowa | Mun. Pwr. | Gen. | Wor | 1 | 300 | 400 | 6 | 10¼ | 14½ | 4 | A | M | 0.40 |
| | | | | | 1 | 225 | 327 | 3 | 13½ | 17½ | 4 | A | M | 0.40 |
| U. S. Dept. of Interior | Gallup, N. M. | Pwr. | Gen. | FM | 1 | 300 | 300 | 4 | 14 | 17 | 2 | A | M | 0.38 |
| | | | | | 1 | 120 | 360 | 2 | 12 | 15 | 2 | A | M | 0.40 |
| Crown Point | Ind. | Mun. Add. | Gen. | FM | 1 | 300 | 300 | 4 | 14 | 17 | 2 | A | M | 0.38 |
| Duluth Univ. Milling Co. | Duluth, Minn. | Pwr. | Shaft | FM | 1 | 300 | 300 | 4 | 14 | 17 | 2 | A | M | 0.38 |
| Hunt Drainage Dist. | Warsaw, Ill. | Pump | Pump | Buc | 1 | 300 | 435 | 8 | 9½ | 14 | 4 | A | M | 0.40 |
| Rockford Screw Prod. Co. | Rockford, Ill. | Pwr. | Gen. | FM | 1 | 300 | 300 | 4 | 14 | 17 | 2 | A | M | 0.38 |
| Central Ice Co. | Brooklyn, N. Y. | Ref. | Comp. | CP | 2 | 288 | 300 | 3 | 14 | 21 | 4 | A | M | 0.40 |
| Corning | Iowa | Lt. & Pwr. | Gen. | FM | 1 | 280 | 300 | 4 | 14 | 17 | 2 | A | M | 0.38 |
| | | | | | 2 | 210 | 300 | 3 | 14 | 17 | 2 | A | M | 0.38 |
| General Electric Co. | Erie, Pa. | Loco. | Gen. | CB | 2 | 275 | 600 | 6 | 8 | 10½ | 4 | A | M | 0.40 |
| C. Black Sand & Gravel Co. | Hammond, La. | Pwr. | Pump | Buc | 1 | 262½ | 400 | 7 | 9½ | 14 | 4 | A | M | 0.40 |
| Gold Hill Mines, Inc. | Shoup, Idaho | Pwr. | Ball Mill & Gen. | Buc | 1 | 262½ | 400 | 7 | 9½ | 14 | 4 | A | M | 0.40 |
| Kivett & Reel, Inc. | Sun, La. | Pwr. | Gen. | Buc | 1 | 262½ | 400 | 7 | 9½ | 14 | 4 | A | M | 0.40 |
| W. S. Kunkle | ..... | Power | Comp. & Gen. | DLV | 1 | 260 | 277 | 4 | 14½ | 18 | 4 | A | M | 0.44 |
| Texas Empire Pipeline Co. | Mattoon, Ill. | Oil | Pump | DLV | 1 | 240 | 600 | 6 | 19 | 11 | 4 | A | M | 0.41 |
| Avondale Farms Dairy | Bethlehem, Pa. | Ref. & Pwr. | Gen. & Comp. | Buc | 1 | 225 | 400 | 6 | 9½ | 14 | 4 | A | M | 0.40 |
| | | | | | 1 | 112½ | 400 | 3 | 9½ | 14 | 4 | A | M | 0.40 |
| | | | | | 1 | 75 | 400 | 2 | 9½ | 14 | 4 | A | M | 0.40 |
| Bowling Green Milling Co. | Bowling Green, Ky. | Flour | Mill | Buc | 1 | 225 | 400 | 6 | 9½ | 14 | 4 | A | M | 0.40 |
| 8th Ave. & 50th St. Bldg. | New York, N. Y. | Plt. Pwr. | Gen. | IR | 1 | 225 | 360 | 4 | 11 | 18 | 4 | A | M | .... |
| G. R. Fibre Cord Co. | Grand Rapids, Mich. | Pwr. | Gen. | Buc | 1 | 225 | 400 | 6 | 9½ | 14 | 4 | A | M | 0.40 |
| La Fourche Shrimp & Oyster Co. | Napoleonville, La. | Ref. & Powr. | Comp & Gen. | Buc | 1 | 225 | 327 | 6 | 9½ | 14 | 4 | A | M | 0.40 |
| Marion Mach. Fdry & Supply Co. | Scottdale, Pa. | Pwr. | Alt. & Gen. | Buc | 1 | 225 | 400 | 6 | 9½ | 14 | 4 | A | M | 0.40 |
| Gaddis Art. Ice Co. | Van Wert, Ohio | Ref. & Pwr. | Gen. & Comp. | Buc | 1 | 225 | 360 | 6 | 9½ | 14 | 4 | A | M | 0.40 |
| Continental Oil Co. | Ponca City, Okla. | Oil | Pump | DLV | 1 | 200 | 1,200 | 6 | 7 | 8 | 4 | A | M | 0.44 |
| Shawnee | Okla. | Pump | Gen. | Wor | 1 | 200 | 400 | 4 | 10¾ | 14½ | 4 | A | M | 0.41 |
| **100-200 Hp. per Engine** | | | | | | | | | | | | | | |
| Korn Leather Co. | Peabody, Mass. | Pwr. | Alt. | Bro | 3 | 187½ | 720 | 6 | 7¼ | 9 5/16 | 4 | A | M | 0.41 |
| Arabi Packing Co. | Arabi, La. | Pwr. | Gen. | Buc | 1 | 187½ | 400 | 5 | 9½ | 14 | 4 | A | M | 0.40 |
| Keynes Bros. Flour Mill | Logan, Ohio | Flour | Mill | Buc | 1 | 187½ | 400 | 5 | 9½ | 14 | 4 | A | M | 0.40 |
| Neon Prod. Inc. | Lima, Ohio | Air & Pwr. | Comp. & Gen. | Buc | 1 | 187½ | 400 | 5 | 9½ | 14 | 4 | A | M | 0.40 |
| Ozark Oak Flooring Co. | Bismarck, Mo. | Pwr. | Gen. | Buc | 1 | 187½ | 400 | 5 | 9½ | 14 | 4 | A | M | 0.40 |
| Tecumseh Gravel Co. | Tecumseh, Mich. | Gravel | Pump | Buc | 1 | 187½ | 400 | 5 | 9½ | 14 | 4 | A | M | 0.40 |

Compiled by E. J. Tangerman
*Associate Editor, POWER*

| Owner | Location | Service | Equipt. Driven | Engine Builder | No. Units | Horsepower Each | Speed R.P.M. | No. Cyl. | Cyl. Bore | Cyl. Stroke | Cycle | Starting Method | Fuel Injection | Fuel Consump. lb. per h. hp. |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Ogden | Iowa | Mun. Pwr. | Gen. | CB | 2 | 185 | 400 | 4 | 10½ | 13½ | 4 | A | M | 0.40 |
| | | | | | 1 | 120 | 400 | 4 | 9 | 12 | 4 | A | M | 0.40 |
| Duncannon | Pa. | Mun. Add. | Gen. | FM | 1 | 180 | 360 | 3 | 12 | 15 | 2 | A | M | 0.40 |
| Amal. Housing Corp. | New York, N. Y. | Lghtg. | Gen. | CB | 3 | 160 | 514 | 6 | 8 | 10½ | 4 | A | M | 0.40 |
| Allegri Ice Co. | Nutley, N. J. | Ref. & Pwr. | Shaft | CP | 1 | 160 | 327 | 4 | 10¾ | 13¾ | 4 | A | M | 0.40 |
| Eldon | Iowa | Mun. Pwr. | Gen. | FM | 3 | 150 | 300 | 2 | 14 | 17 | 2 | A | M | 0.38 |
| Havana Ice Co. | Havana, Fla. | Pwr. | Gen. | Buc | 1 | 150 | 400 | 4 | 9½ | 14 | 4 | A | M | 0.40 |
| Imperial Metal Products Co. | Grand Rapids, Mich. | Pwr. | Gen. | Buc | 1 | 150 | 400 | 4 | 9½ | 14 | 4 | A | M | 0.40 |
| Kuenzel Mills | New Bremen, Ohio | Flr. & Wln. | Mill | Buc | 1 | 150 | 400 | 4 | 9½ | 14 | 4 | A | M | 0.40 |
| Lincoln Warehouse Corp. | New York, N. Y. | Pwr. | Gen. | CP | 1 | 150 | 400 | 3 | 10¾ | 14½ | 4 | E | M | 0.40 |
| McDonald Ice Co. | McDonald, Pa. | Ref. & Pwr. | Comp. & Gen. | Buc | 1 | 150 | 225 | 4 | 9½ | 14 | 4 | A | M | 0.40 |
| J. P. Millbank | Chillicothe, Miss. | Flour | Mill | Buc | 1 | 150 | 400 | 4 | 9½ | 14 | 4 | A | M | 0.40 |
| Reidsville Ice & Coal Co. | Reidsville, N. C. | Ref. & Pwr. | Gen. & Comp. | Buc | 1 | 150 | 400 | 4 | 9½ | 14 | 4 | A | M | 0.40 |
| Hiram | Ohio | Mun. Pwr. | Alt. | And | 1 | 140 | 300 | 2 | 14 | 18 | 2 | A | M | .... |
| | | | | | 2 | 120 | 300 | 2 | 12½ | 18 | 2 | A | M | .... |
| Bernards Water Co. | Bernardsville, N. J. | Wat. Sup. | Pump | CB | 1 | 135 | 400 | 4 | 9 | 12 | 4 | A | M | 0.40 |
| Royal Amer. Shows | Miami, Fla. | Lt. | Gen. | Cat | 7 | 125 | 850 | 6 | 5½ | 8 | 4 | G | M | 0.43 |
| Oranogo Mutual Mining Co. | Oranogo, Calif. | Air | Comp. | Cat | 3 | 125 | 850 | 6 | 5¼ | 8 | 4 | G | M | 0.43 |
| Callahan Gunther & Shirley | Winterhaven, Calif. | Lt. | Gen. | Cat | 2 | 125 | 850 | 6 | 5¼ | 8 | 4 | G | M | 0.43 |
| | | | | | 3 | 102 | 850 | 6 | 5¼ | 8 | 4 | G | M | 0.43 |
| Academy Gardens | New York, N. Y. | Apt. Pwr. | Gen. | Wor | 2 | 125 | 514 | 5 | 8 | 10½ | 4 | A | M | 0.41 |
| C. T. Welch, Inc. | Quincy, Ill. | Towboat | Prop. | Cat | 2 | 125 | 850 | 6 | 5¼ | 8 | 4 | G | M | 0.43 |
| Fay Placer Mine | Lincoln, Cal. | Pwr. | Shovel | Cat | 1 | 125 | 850 | 6 | 5¼ | 8 | 4 | G | M | 0.43 |
| | | Pwr. | Convey. | Cat | 1 | 77 | 850 | 4 | 5¾ | 8 | 4 | G | M | 0.43 |
| E. Eggers | Houston, Tex. | Tug | Prop. | Cat | 1 | 125 | 850 | 6 | 5¼ | 8 | 4 | G | M | 0.43 |
| Renwick | Iowa | Mun. Pwr. | Gen. | Wor | 1 | 125 | 514 | 5 | 8 | 10½ | 4 | A | M | .... |
| Liberty Brass Turning Co. | New York, N. Y. | Pwr. Lt. | Gen. | Bol | 1 | 120 | 400 | 2 | 13 | 13½ | 2 | A | M | .... |
| U. S. Treas. Dept. | New York, N. Y. | Pool | Pump | CB | 1 | 120 | 350 | 3 | 10½ | 13½ | 4 | A | M | 0.40 |
| Aaron Mills | Martinsville, Va. | Flour | Mill | Buc | 1 | 112½ | 400 | 3 | 9½ | 14 | 4 | A | M | 0.40 |
| Farmers Gin Co. | Darling, Miss. | Pwr. | Gin | Buc | 1 | 112½ | 400 | 3 | 9½ | 14 | 4 | A | M | 0.40 |
| Farmers Gin Co. | Coahoma, Miss. | Pwr. | Gin | Buc | 1 | 112½ | 400 | 3 | 9½ | 14 | 4 | A | M | 0.40 |
| Farmers Gin Co. | Tutwiler, Miss. | Pwr. | Gin | Buc | 1 | 112½ | 400 | 3 | 9½ | 14 | 4 | A | M | 0.40 |
| Pamlico Sea Foods | Vandemere, N. C. | Ref. & Pwr. | Gen. & Comp. | Buc | 1 | 112½ | 360 | 3 | 9½ | 14 | 4 | A | M | 0.40 |
| Layne La. Co. | Lake Chas., La. | Well. | Pump | CB | 2 | 110H | 327 | 2T | 11½ | 15 | 2 | A | M | 0.40 |
| Oil Well Supply Co. | Houston, Tex. | Oil well | Pump | CB | 1 | 110H | 327 | 2T | 11½ | 15 | 2 | A | M | 0.40 |
| A. B. Brown Co. | Queens, N. Y. | Plant Pwr. | Gen. | Bol | 1 | 100 | 900 | 4 | 7⅛ | 7⅛ | 2 | A | M | .... |
| Colussy Ice Co. | Bridge Villa, Pa. | Pwr. | Gen. | Buc | 1 | 100 | 514 | 5 | 9½ | 14 | 4 | A | M | 0.40 |
| **50-100 Hp. per Engine** | | | | | | | | | | | | | | |
| American Die Casting Corp. | L. I. City N. Y. | Pwr. & Lt. | Gen. | Bol | 1 | 90 | 425 | 2 | 11½ | 12 1/16 | 2 | A | M | .... |
| Cherry Growers Packing Co. | Traverse City, Mich. | Ref. | Comp. | Bol | 1 | 90 | 425 | 2 | 11½ | 12 1/16 | 2 | A | M | .... |
| Manhattan College | New York, N. Y. | Pwr. | Alt. | FM | 3 | 80 | 1,200 | 4 | 5½ | 7½ | 4 | A | M | .... |
| G. M. Anthony | Straustown, Pa. | Flr. Mill | Shaft | Buc | 1 | 80 | 514 | 4 | 9½ | 14 | 4 | A | M | 0.40 |
| Cincinnati Tool Co. | Cincinnati, Ohio | Pwr. | Gen. & Shaft | Buc | 1 | 80 | 400 | 2 | 9½ | 14 | 4 | A | M | 0.40 |
| Statesville Ice Co. | Statesville, N. C. | Ref. | Comp. | Buc | 1 | 80 | 514 | 4 | 9½ | 14 | 4 | A | M | 0.40 |
| W. Cairo Elev. Co. | Cairo, Ohio | Elev. | Gen. | Buc | 1 | 80 | 514 | 4 | 9½ | 14 | 4 | A | M | 0.40 |
| M-W-A-K Co. | Grand Coulee, Wis. | Loco. | Gen. | Cat | 14 | 77 | 850 | 4 | 5½ | 8 | 4 | G | M | 0.43 |
| Mexican Gov. | Mexico | Rd. Build | Crush | Cat | 12 | 77 | 850 | 4 | 5½ | 8 | 4 | G | M | 0.43 |
| Ideal Clamp Mfg. Co. | Brooklyn, N. Y. | Plt. Pwr. | Gen. | CB | 1 | 75 | 450 | 3 | 7½ | 10½ | 4 | A | M | 0.40 |
| Texas Co. | Los Angeles, Cal. | Oil | Pump | Sup | 2 | 72 | 1,200 | 6 | 4½ | 5¼ | 4 | E | M | 0.45 |
| Alderfer Bol. Co. | Harleysville, Pa. | Pwr. | Gen. | Sup | 1 | 72 | 1,200 | 6 | 4½ | 5¼ | 4 | E | M | 0.45 |
| American Garage | Piscataway, N. J. | Lt. | Gen. | Sup | 1 | 72 | 1,200 | 6 | 4½ | 5¼ | 4 | E | M | 0.45 |
| Amer. Tankers Corp. of Del. | Mobile, Ala. | Air | Comp. | Sup | 1 | 72 | 1,200 | 6 | 4½ | 5¼ | 4 | E | M | 0.45 |
| Yacht "Janidore" | San Francisco, Cal. | Lt. | Gen. | Sup | 1 | 72 | 1,200 | 6 | 4½ | 5¼ | 4 | E | M | 0.45 |
| M. M. Nolt & Son | New Holland, Pa. | Ref. | Comp. | Sup | 1 | 72 | 1,200 | 6 | 4½ | 5¼ | 4 | E | M | 0.45 |
| Otto Engine Works | Holmesburg, Pa. | Pwr. | Gen. | Sup | 1 | 72 | 1,200 | 6 | 4½ | 5¼ | 4 | E | M | 0.45 |
| Smith, Levin & Harris, Inc. | Lopez, Pa. | Lt. & Pwr. | Gen. | Sup | 1 | 72 | 1,200 | 6 | 4½ | 5¼ | 4 | E | M | 0.45 |
| Spadaro Constr. Co. | Moore Haven, Fla. | Water | Pump | Sup | 1 | 72 | 1,200 | 6 | 4½ | 5¼ | 4 | E | M | 0.45 |
| Stark Elec. Co. | Stewart, Fla. | Lt. | Gen. | Sup | 1 | 72 | 1,200 | 6 | 4½ | 5¼ | 4 | E | M | 0.45 |
| Wheeler Shipyard | Bklyn, N. Y. | Pwr. Add. | Gen. | Sup | 1 | 72 | 1,200 | 6 | 4½ | 5¼ | 4 | E | M | 0.45 |
| K & J Market | Newark, N. J. | Lt. | Gen. | Bol | 1 | 70 | 450 | 2 | 10½ | 11 | 2 | A | M | .... |
| J. H. Richardson | St. Joseph, Tenn. | Gin | Shaft | Bol | 1 | 70 | 450 | 2 | 10½ | 11 | 2 | A | M | .... |
| Pan-Amer. Airways | Wake & Midway Is. | Lt. & Pwr. | Gen. | Cat | 4 | 63 | 850 | 4 | 5½ | 8 | 4 | G | M | 0.43 |
| T. P. & W. R.R. | Peoria, Ill. | Ref. | Comp. | Cat | 2 | 63 | 850 | 4 | 5½ | 8 | 4 | G | M | 0.43 |
| Havana Ice Co. | Tallahassee, Fla. | Pwr. | Shaft. | Buc | 1 | 60 | 514 | 3 | 9½ | 14 | 4 | A | M | 0.40 |
| Shawnee & New Straitsville Ice & Ice-Cream Co. | Shawnee, Ohio | Ref. | Comp. | Buc | 1 | 60 | 514 | 3 | 9½ | 14 | 4 | A | M | 0.40 |
| Clover Hill Dairy | Lunenbury, Mass. | Lt. & Pwr. | Gen. | Rus | 1 | 57 | 275 | 1 | 11½ | 21 | 4 | A | M | 0.40 |
| | | Rest. Lt. | Gen. | Lis | 1 | 38 | 1,000 | 4 | 4½ | 5½ | 4 | A | M | 0.42 |
| | | Lt. & Pwr. | Gen. | Rus | 1 | 34 | 310 | .. | 9½ | 17½ | 4 | A | M | 0.42 |
| | | Rest. Lt. | Gen. | Lis | 1 | 27 | 1,000 | 3 | 4½ | 5½ | 4 | H | M | 0.42 |
| | | Rest. Lt. | Gen. | Lis | 1 | 5 | 600 | 1 | 4½ | 5½ | 4 | H | M | 0.46 |
| Garden City Garage | Garden City, N. Y. | Lt. & Pwr. | Gen. | Bol | 1 | 50 | 800 | 2 | 7⅛ | 7⅛ | 2 | A | M | .... |
| Red Lion Stores | Queens, N. Y. | Lt. | Gen. | Bol | 1 | 50 | 800 | 2 | 7⅛ | 7⅛ | 2 | A | M | .... |
| Tobins Drug Store | New Brunswick, N. J. | Lt. | Gen. | Bol | 1 | 50 | 800 | 2 | 7⅛ | 7⅛ | 2 | A | M | .... |
| **Under 50 Hp. per Engine** | | | | | | | | | | | | | | |
| Kent's Rest. & Bak. Co. | Atlantic City, N. J. | Ref. | Comp. | Sup | 1 | 48 | 1,200 | 4 | 4½ | 5½ | 4 | E | M | 0.45 |
| Sunbury Milk Prod. | Sunbury, Pa. | Pwr. | Gen. | Sup | 1 | 48 | 1,200 | 4 | 4½ | 5½ | 4 | E | M | 0.45 |
| Johnson & Ashman | Freeland, Pa. | Ref. | Comp. | Rus | 1 | 47 | 290 | 1 | 10½ | 19½ | 4 | A | M | 0.42 |
| Trenton Packing Co. | Trenton, N. J. | Ref. | Comp. | Rus | 1 | 47 | 290 | 1 | 10½ | 19½ | 4 | A | M | 0.42 |
| Sam's Diner | Springfield, Mass. | Lt. & Pwr. | Gen. | Bol | 1 | 45 | 425 | 1 | 11¼ | 12 3/16 | 2 | A | M | .... |
| Brevoort Garage | New York, N. Y. | Lt. & Pwr. | Gen. | Lis | 2 | 38 | 1,000 | 4 | 4½ | 5½ | 4 | H | M | 0.44 |
| Point O'Woods Assn. | L. I., N. Y. | Mun. Lt. | Alt. | Lis | 1 | 38 | 1,000 | 4 | 4½ | 5½ | 4 | H | M | 0.44 |
| Dutchland Farms | W. Springfield, Mass. | Pwr. | Gen. | Rus | 1 | 34 | 310 | 1 | 9½ | 17½ | 4 | A | M | 0.42 |
| Turnbull Ice Cream | Greenfield, Mass. | Ref. | Comp. | Rus | 1 | 34 | 310 | 1 | 9½ | 17½ | 4 | A | M | 0.42 |
| West View Dairy | Souderton, Pa. | Lt. & Pwr. | Gen. | Rus | 1 | 34 | 310 | 1 | 9½ | 17½ | 4 | A | M | 0.42 |
| Morrissey Bros. | N. Adams, Mass. | Rest. Lt. | Gen. | Rus | 1 | 28 | 320 | 1 | 8½ | 16½ | 4 | A | M | 0.44 |
| | | | | Lis | 1 | 10 | 600 | 2 | 4½ | 5½ | 4 | H | M | 0.46 |
| Roth Packing Co. | Union City, N. J. | Ref. & Pwr. | Gen. | Lis | 2 | 27 | 1,000 | 3 | 4½ | 5½ | 4 | H | M | 0.42 |
| Colony Club | New York, N. Y. | Ref. | Comp. | Lis | 1 | 27 | 1,000 | 3 | 4½ | 5½ | 4 | H | M | 0.42 |
| Port Chester | N. Y. | Sew. Disp. | Gen. | Lis | 1 | 27 | 1,000 | 3 | 4½ | 5½ | 4 | H | M | 0.42 |
| Yonkers Fruit Co. | Yonkers, N. Y. | Ref. | Comp. | Lis | 1 | 27 | 1,000 | 3 | 4½ | 5½ | 4 | H | M | 0.42 |
| Henry Huss | Lakewood, N. J. | Lt. | Gen. | Bol | 1 | 25 | 800 | 1 | 7⅛ | 7⅛ | 2 | A | M | .... |
| Standard Oil Co. | Caripita, Venz. | Lt. | Gen. | Sup | 3 | 24 | 1,200 | 2 | 4½ | 5½ | 4 | E | M | 0.45 |
| A. M. Becker | Vineland, N. J. | Pwr. | Gen. | Sup | 1 | 24 | 1,200 | 2 | 4½ | 5½ | 4 | E | M | 0.45 |
| Boulevard Rest. | Camden, N. J. | Lt. & Pwr. | Gen. | Sup | 1 | 24 | 1,200 | 2 | 4½ | 5½ | 4 | E | M | 0.45 |
| V. S. Budny | Trenton, N. J. | Lt. & Pwr. | Gen. | Sup | 1 | 24 | 1,200 | 2 | 4½ | 5½ | 4 | E | M | 0.45 |
| Lincoln Creamery | Asbury Park, N. J. | Ref. | Comp. | Bol | 1 | 20 | 900 | 2 | 4¼ | 4¼ | 2 | H | M | .... |
| James Mix | Suffield, Conn. | Lt. | Gen. | Bol | 1 | 20 | 900 | 2 | 4¼ | 4¼ | 2 | H | M | .... |
| R. M. Burtis | Viola, N. Y. | Ref. | Comp. | Lis | 1 | 18 | 1,000 | 2 | 4½ | 5½ | 4 | H | M | 0.44 |
| John Villano | Waterbury, Conn. | Pwr. | Gen. | Lis | 1 | 14 | 1,000 | 2 | 4½ | 4½ | 4 | H | M | 0.48 |
| Atkinson Dredging | Norfolk, Va. | Lt. | Gen. | Bol | 1 | 10 | 900 | 1 | 4¾ | 4¾ | 2 | H | M | .... |
| Federal Industrial Corp. | Paterson, N. J. | Lt. | Gen. | Bol | 1 | 10 | 900 | 1 | 4¾ | 4¾ | 2 | H | M | .... |
| Keller Market | Plainfield, N. J. | Ref. | Comp. | Bol | 1 | 10 | 900 | 1 | 4¾ | 4¾ | 2 | H | M | .... |
| North Adams Garage | North Adams, Mass. | Lt. | Gen. | Bol | 1 | 10 | 900 | 1 | 4¾ | 4¾ | 2 | H | M | .... |
| R. J. Balliet | Slatedale, Pa. | Pwr. | Gen. | Lis | 1 | 9 | 1,000 | 1 | 4½ | 5½ | 4 | H | M | 0.46 |
| Pearl River Recreation | N. Y. | Lt. | Gen. | Lis | 1 | 9 | 1,000 | 1 | 4½ | 5½ | 4 | H | M | 0.46 |
| Annex Dairy, Inc. | New Haven, Conn. | Ref. | Comp. | Lis | 1 | 8 | 1,000 | 1 | 4½ | 4½ | 4 | H | M | 0.48 |
| J. W. H. Behrens | E. Mauch Chunk, Pa. | Lt. | Gen. | Lis | 1 | 8 | 1,200 | 1 | 4½ | 4½ | 4 | H | M | 0.48 |
| N. Y. Diesel Inst. | Albany, N. Y. | Lt. | Gen. | Lis | 1 | 8 | 1,200 | 1 | 4½ | 4½ | 4 | H | M | 0.48 |
| H. Van Dorp | Terra Ceia, N. C. | Ref. | Comp. | Lis | 1 | 8 | 1,200 | 1 | 4½ | 4½ | 4 | H | M | 0.48 |
| Vernon H. Brown | Port Jervis, N. Y. | Hse. Lt. | Gen. | Lis | 1 | 5 | 600 | 1 | 4½ | 5½ | 4 | H | M | 0.45 |

**Abbreviations**—AI—Atlas Imperial, And—Anderson, Bro—Bromfield-Deutz, Bol—Bolinders, Buc—Buckeye, Bud—Buda, BuS—Busch-Sulzer, Cat—Caterpillar, DLV—DeLaVergne, FM—Fairbanks Morse, CB—Cooper Bessemer, Ful—Fulton, HOR—Hooven, Owens, Rentschler, IR—Ingersoll-Rand, Lis—Lister, McI—McIntosh and Seymour, Nor—Nordberg, Pac—Pacific, Rus—Ruston, Sup—Superior, Win—Winton, Wor—Worthington, Wau—Waukesha. Add.—addition, Alt.—alternator, C. S.—central station, Gen.—generator, H—horizontal Irrig—irrigation, Lt.—light, Mun.—municipal, Plt—plant. Pwr.—power, Ref.—refrigeration, Wtr.—water.

Under fuel injection—M is mechanical, A is air. Under starting, A is air, H hand, E electric, G gasoline.

# GAS ENGINES

TWENTY years ago engineers agreed that the gas engine was on the way out, except where process gas was available, as in steel mills. But new supplies of natural gas were tapped and over 80,000 miles of pipelines spread through 38 states (exceptions are the 6 New England States, N. C., S. C., Nev., and Idaho), until today we find a 26-in. line running into Chicago and a 22-in. line under construction to Detroit. With distribution piping, the total is 165,000 miles. Natural gas from Pennsylvania is piped to just across the Hudson from New York. Pipelines themselves use gas engines.

Refineries have also discovered that still gas provides more power in a gas engine than under boilers. Steel mills add waste-heat boilers to their big horizontal engines and increase the useful heat from the fuel to 70% total. Sewage-disposal plants find that a small vertical engine, powered by sludge gas, will give them sufficient power at a fuel cost of practically nothing. Oil-well drillers power their engines with oil until the well comes in, then shift over to the gas that comes with the oil. Plants along the pipelines put in engines which can be converted from gas engine to diesel in a few hours depending upon the economics of fuel supply. Manufactured-gas plants mix natural with their gas, using gas-engines for boosters, or use them for distribution.

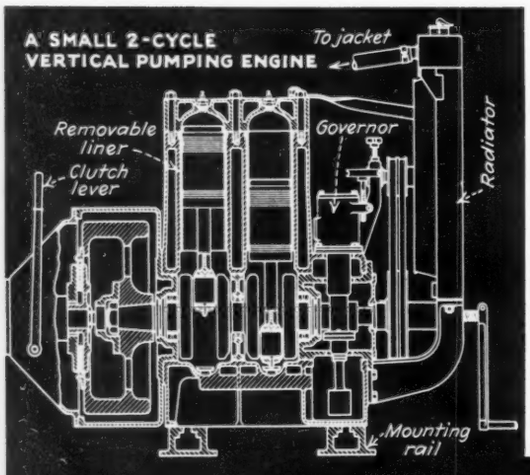
Large gas engines above 400 hp. are 4-cycle with full-load fuel economy of 10,000 B.t.u. per b.-hp.-hr. (10 in. or larger bore—small engines increase this figure 5 to 15%). Below 400 hp. the 2-cycle is gaining ground because of recent progress toward improved scavenging. By injecting the fuel under 12 to 15 lb. pressure after the air has been injected and the ports closed, 2-cycle consumption has been cut from 13,000 B.t.u. per b.-hp.-hr. to approximately the same figure as that for 4-cycle engines. (These figures compare with 7,800 B.t.u. per hp.-hr. for the diesel.) Up to about 250 hp., several times as many 2-cycle engines as 4-cycle have been installed in recent years.

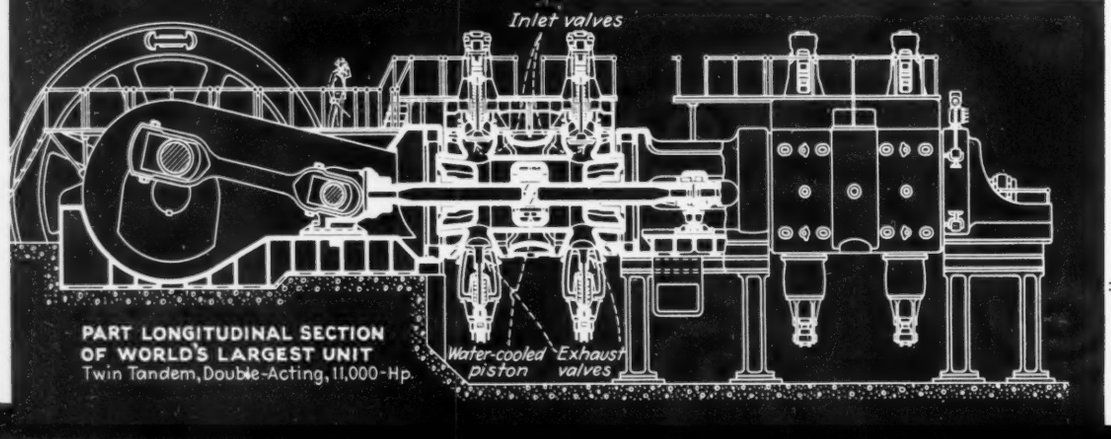
Normally, large gas engines are double acting for saving in weight, cost per horsepower and floor space. Double-acting designs are much more readily produced in the gas engine than in the diesel because greater clearance volume makes possible the installation of inlet and exhaust valves of proper size without cramping or unduly distorting clearance space.

Normally, these engines are horizontal rather than vertical for accessibility and for adaption to cylinders in tandem with simple valve gear, there being only one gear-driven crank or eccentric shaft. In fact, except in very small sizes, tandem cylinders are necessary to gain full advantage of the horizontal, double-acting type. These engines are primarily designed for natural-gas compression, so this is another reason for their common horizontal construction. The tandem has one crank and connecting rod, one main frame and one main and one outboard bearing, yet is equivalent to four single-acting cylinders except in smoothness. The twin tandem doubles the number of connecting rods, cranks and main frames, but uses only two main bearings and is similarly equivalent to eight single-acting cylinders. Fewer moving parts also means less maintenance and less noise. Operators can reach almost any part from the engine-room floor and can quickly detect any failure in oil supply, a sticking valve, or a hot bearing. Alignment need not be so exact, oil consumption is lower and the oil requires changing or cleaning less often because crankcase oil is not contaminated. All important torsional critical speeds are far above operating speeds.

Horizontal engines require greater floor space and foundation than verticals, inertia of reciprocating parts is high and acts horizontally (although this avoids torsional vibration troubles inherent in the vertical engine) and the engine cannot be balanced as well as some vertical cylinder combinations. Foundation mass must be greater.

The horizontal slow-speed engine is ideal for connection to direct-connected reciprocating compressors, a standard opposed-type connection giving fewest parts and minimum floor space. But the vertical is far better for connection to electric generators normally because the speed is higher and it avoids the necessity for belted exciters. The horizontal tandem, due to its large reciprocating parts, is seldom run over 130 r.p.m. on compressor drives and 150




PART LONGITUDINAL SECTION OF WORLD'S LARGEST UNIT
Twin Tandem, Double-Acting, 11,000-Hp.

Cross-sections of the world's largest gas engine, a horizontal, double-acting twin-tandem 4-cycle, 6,600-kw. engine-compressor at the So. Chicago works of Illinois Steel Co., and a 40-hp., 2-cycle vertical with built-in clutch, radiator, fan and water pump. Man and crank indicate comparative size

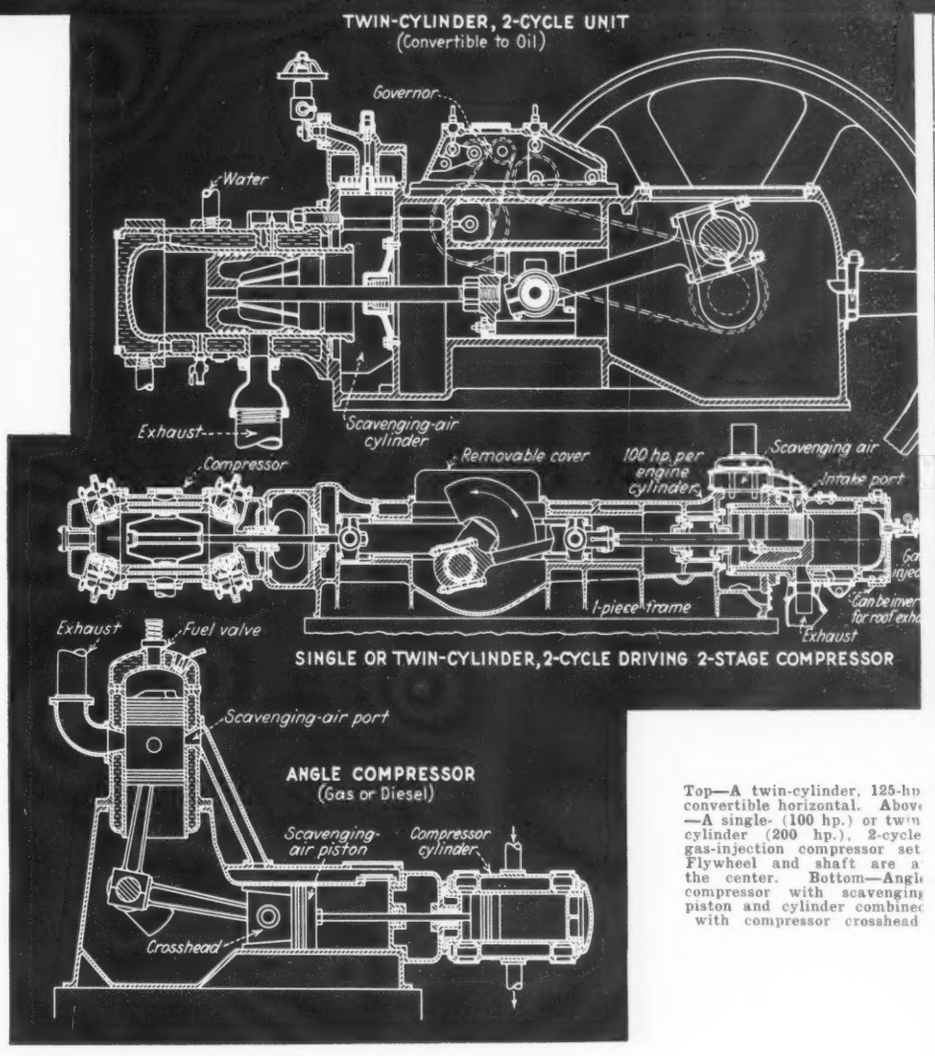
to 160 r.p.m. on generator drives, while the vertical may run as high as 300 or 400 in an equivalent size.

The largest vertical engines thus far built in this country are 1,800-hp., 225-r.p.m. units. These 12-cyl. engines have 6 cyl. each side of the generator and reach an economy of 9,700 B.t.u. per b.-hp.-hr.

High-speed (900- to 1,200-r.p.m.) vertical gas engines are a composite of automotive and diesel designs. They normally are 4-cycle single-acting with double overhead exhaust and inlet poppet valves. Removable liners, trunk pistons, automotive-type connecting rods, full-pressure lubrication, oil filters and coolers, crankcase oil pumps, etc., are standard. Water jackets may be individual, or cylinders may be jacketed in groups of 2, 4, 6 or 8. One design replaces the jacket with an open tank. Usually block frame and head are cast iron, for weight saving is not particularly important. Piston rod, camshaft and crankshaft are drop-forged, and pistons may be cast iron or aluminum alloy. Piston rod may be a cap type with shims, or may have adjusting wedges like a steam engine. Mixing valves are provided for each cylinder pair. Oil contamination is avoided by wiper rings on piston skirts. Jacket water may be circulated at higher velocity at liner tops, and liner supporting shoulders are moved down, both to increase the rate of heat transfer.

Smaller engines, both vertical and horizontal, go down to almost any desired minimum. They usually have integral radiators and lubricating systems just like gasoline engines. In fact, almost any gas engine can be converted for gasoline operation and vice versa, simply by exchanging the mixing valve for a carburetor.

Some years ago, a gas pump for pumping water by displacement was introduced. More modern types are 2-cycle and much simplified, increasing capacity 66% and reducing sensitiveness to changes in sump level and valve-spring tension (See *Power*, Mid-December, 1933, page 696). The pump has no crankshaft, piston or connecting rod, these functions being performed by a water column. Compression pressure is about 30 lb. per sq.in., combustion peak about 140. Pump discharge is about 8 or 10 times the depth of the gas charge, due to the inertia effect of the water column.




Top—A twin-cylinder, 125-hp convertible horizontal. Above —A single- (100 hp.) or twin cylinder (200 hp.), 2-cycle gas-injection compressor set. Flywheel and shaft are at the center. Bottom—Angle compressor with scavenging piston and cylinder combined with compressor crosshead

Interesting is the combined vertical engine-compressor in which one or two engine cylinders are replaced by compressor cylinders. Design is worked out so that the compressor cylinder or cylinders are at the flywheel end of the machine, utilizing the same base, crankshaft, and cylinder block but a special head.

Piston speeds up to 900 or 1,000 f.p.m. are permissible for general power work, but for compressor service, due to limitation of compressor valve area and inertia of reciprocating weights, 750 f.p.m. is standard in larger sizes, slightly lower in smaller ones.

Very large horizontals may use 110-volt d.c. for ignition, magneto, or battery. Smaller engines use magneto or battery. Battery ignition is replacing mechanical make-and-break or the magneto, because in compression service and consequent slow speeds the magneto loses efficiency. High-tension distributors are somewhat better, but there is still the danger from running high-tension leads over the engine. Most satisfactory is the low-tension distributor and storage battery with a motor-generator set floating on the line. The distributor provides dual ignition and permits individual adjustment of each breaker. The distributor can be advanced or retarded and breakers removed from the distributor head while the engine is running. Usually separate ignition is provided for each plug in a multi-plug cylinder.

Fuel economy for horizontal and vertical engines runs about the same, about 12 cu.ft. of 1,000 B.t.u. gas for small and medium-size engines, dropping to 10 in larger ones. A 2-cycle engine without fuel injection requires about 14 cu.ft. For 530-B.t.u. gas, the comparable figures are 26 and 23 cu.ft. According to several gas-engine manufacturers, a gas engine running on 1,000-B.t.u. gas costing 35 cents per 1,000 cu.ft. compares favorably with a diesel running on 5-cent (per gal.) oil. The gas engine has explosion pressures only about half those of the diesel, permitting lighter construction and reducing maintenance costs. A safe figure for maintenance on slow-speed engines is 2% of the first cost per 8,000-hr. operating year, rising to 3% for medium speed and 5% for

high speed. Large tandem double-acting engines require much less maintenance than do trunk-piston verticals, because the piston is "floated," i.e., has crossheads outside the cylinder at each end. In one examination of more than 200 21½ x 36-in. double-acting engine cylinders, most of them running 24 hr. a day at 100 to 125 r.p.m., average wear on the 21½-in. diameter was a little under 0.002 in. per year!

The gas engine is as flexible to load changes as a steam engine or a diesel, and as capable of handling an overload as a diesel, if properly rated. Some recent engines are equipped with superchargers to handle overloads or to compensate for altitude. Supercharging is usually practical only on larger engines. Considering sea level as 100%, an elevation of 1,000 ft. reduces power delivered to 97%, 2,000 ft. reduces it to 92%, 3,000 to 88%, 5,000 to 82%, 7,500 to 73.5%, and 10,000 to 66%. Most modern engines, however, can carry sea-level ratings to 5,000 ft. by supercharging, although mean effective pressures are usually held to 80 to 100 lb. per sq.in. with uncooled pistons, and 115 to 130 lb. with cooled pistons. In large engines, most operators prefer around 70 lb. in an engine without water-cooled pistons and exhaust valves.

Gas engines are almost automatic in operation. Compression, peak, terminal and back pressures, ignition and valve timing, mixtures and prematures can all be shown with the ordinary indicator (see July, 1936, *Power* Data Sheet) on all except very high-speed engines. The rig is built in. Peak pressures should be kept as low as possible without a loss in m.e.p. Excessive peaks abuse engine parts, and resulting high temperatures cause greater heat transfer to jackets, hence greater power loss. Early timing or prematures cause high peaks. Terminal pressure must of course be kept as low as possible, otherwise latent expansive power is lost in the exhaust. High terminal pressures are caused by late ignition or a too-rich mixture. Cooling water should normally be held below 140 deg. F. Assuming 110 outlet and 70-deg. inlet temperatures, the temperature rise is 40 deg. Dividing the constant 500 by this rise gives 12.5 gal. of water per b.-hp.-hr. required. As in modern diesel practice, it is advisable to use a closed cooling system when possible, particularly when available cooling water is hard and likely to lime the water jackets. About 3 ft. of 2-in. pipe per b.-hp. is a liberal allowance for cooling surface.

Lubricating oil consumed by a gas engine varies with several factors, whether or not it is reclaimed, type of engine, care and skill of operator. It is safe to figure 2,000 rated hp.-hr. per gal. although 3,000 rated hp.-hr. are now obtained with force-feed systems.

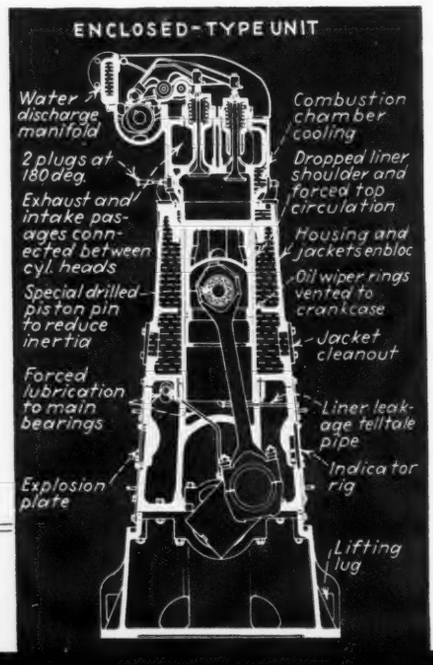
Gas engines lend themselves particularly to automatic control and operation. Electrical means are now available to signal an operator or shut down the engine in case of excessive cooling water temperatures, overheating of a bearing, failure of lubricating oil supply, excessive exhaust temperatures or mechanical damage to any part. Governors of the isochronous type can be applied to maintain constant frequency over wide load fluctuations. Automatic electric relays can be tied in with a battery to open the fuel valve, turn on ignition and crank the engine electrically, as well as to cut out ignition and shut off fuel at once in case of trouble. This would indicate gas engines as standby units in hospitals, continuous-process plants, civic buildings, police and fire headquarters, traffic-control systems, etc., where constant power supply is essential. Batteries can be installed to carry the load during the very short starting periods.

The gas engine's twin brother, the stationary gasoline engine, also finds an important place for such uses. While fuel cost is so high a gasoline engine cannot normally compete with other prime movers, in emergency installations the fuel cost is not so important as low first and installation costs.

The ideal arrangement for a natural-gas pipeline company is to have constant load on its lines. Obviously, however, during the summer non-heating months natural gas load drops off and reaches its peak in the winter when gas is used for heating. In order to maintain load on the line many companies are now offering gas during the summer at low rates. This makes an attractive field for the convertible engine which can be run on gas during the summer when off-peak rates are maintained, then shifted to diesel in the fall when the rate schedule is changed. Of course, the convertible engine may not be as efficient as the straight gas engine or the straight diesel engine because of the differences in explosion load. In a diesel, explosion load runs 550 to 600 lb. per sq.in. while in the gas engine it is only 350 lb. This means that a diesel of a given size must be heavier and stronger. Unless, in conversion, the larger piston and cylinder of a gas engine are changed for a smaller diesel piston and liner at the same time compression is increased, the engine will not make full use of its metal. Of course, this possible inefficiency must be balanced against rate structure and possible lower first cost due to increased production by the engine manufacturer.

For gas-engine installations at sewage-disposal plants, the following figures have been worked out: For each cu.ft. of sludge gas trapped and used per day, you can afford to spend $1 in gas-engine equipment and expect to amortize the investment in ten years, with electric current at 1 cent per kw.-hr. Gas potentially available at an Imhoff tank and sprinkling filter plant is usually more than enough to generate all the power necessary to operate a plant. If the installation is an activated-sludge arrangement, potential generation is only ¾ that needed for operation.

Almost any gas can be used efficiently in a gas engine. Principal requirements are that the gas be supplied steadily at proper pressure to the engine and that it is properly mixed with air in correct ratio. The designer must proportion mixing valve and adjacent parts properly, and the operator must maintain proper adjustments and be sure his regulators are set to give

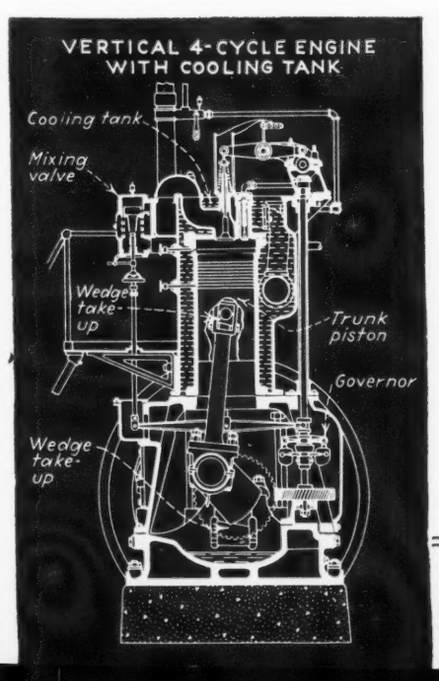

ENCLOSED-TYPE UNIT
Water discharge manifold
2 plugs at 180 deg.
Exhaust and intake passages connected between cyl. heads
Special drilled piston pin to reduce inertia
Forced lubrication to main bearings
Explosion plate
Combustion chamber cooling
Dropped liner shoulder and forced top circulation
Housing and jackets enbloc
Oil wiper rings vented to crankcase
Jacket cleanout
Liner leakage telltale pipe
Indicator rig
Lifting lug

# 52 TYPICAL NEW GAS-ENGINE PLANTS

## 163 Engines Installed Since June, 1934

| Owner | Location | Service | Equipt. Driven | Engine Builder | No. Engines | Horsepower | Speed R.P.M. | No. Cyl. | Cyl. Bore | Cyl. Stroke | Horizontal or Vertical | Single or Tandem | Cycle | B.t.u. per B.Hp.-Hr. | Type of Fuel | B.t.u. per Cu.ft. of Fuel |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Panhandle Eastern Pipe Line Co. | Houstonia, Mo. | Gas | Comp. | CB | 4 | 1,300 | 125 | 4 | 24 | 36 | H | TT | 4 | 10,000 | Nat. | 1,000 |
| | Centralia, Mo. | Gas | Comp. | Wor | 3 | 1,300 | 125 | 4 | 24 | 36 | H | TT | 4 | 10,000 | Nat. | 1,000 |
| | Greensburg, Kans. | Gas | Comp. | CB | 3 | 1,300 | 125 | 4 | 24 | 36 | H | TT | 4 | 10,000 | Nat. | 1,000 |
| | Haven, Kansas | Gas | Comp. | CB | 3 | 1,300 | 125 | 4 | 24 | 36 | H | TT | 4 | 10,000 | Nat. | 1,000 |
| | Olpe, Kan. | Gas | Comp. | Wor | 3 | 1,300 | 125 | 4 | 24 | 36 | H | TT | 4 | 10,000 | Nat. | 1,000 |
| | Pleasant Hill, Ill. | Gas | Comp. | Wor | 2 | 1,300 | 125 | 4 | 24 | 36 | H | TT | 4 | 10,000 | Nat. | 1,000 |
| | Stinnett, Texas | Gas | Comp. | CB | 2 | 1,300 | 125 | 4 | 24 | 36 | H | TT | 4 | 10,000 | Nat. | 1,000 |
| Corning Glass Co. | Corning, N. Y. | Pwr. Add. | Gen. | IR | 1 | 1,200 | 225 | 6 | 22 | 26 | V | S | 4 | 10,000 | Nat. | 1,000 |
| Washington | D.C. | Mun. Sew. | Gen. | RJ | 1 | 1,200 | 225 | 6 | 22 | 26 | V | S | 4 | 8,400 | Slu. | 600 |
| Warren Petroleum Co. | Tulsa, Okla. | Pipeline | Comp. | CB | 8 | 1,000 | 125 | 4 | 21½ | 36 | H | TT | 4 | 10,000 | Nat. | 1,000 |
| Panhandle Eastern Pipe Line Co. | Liberal, Kan. | Gas Add. | Comp. | CB | 2 | 1,000 | 125 | 4 | 21½ | 36 | H | TT | 4 | 10,000 | Nat. | 1,000 |
| | | Pwr. Add. | Gen. | CB | 6 | 200 | 400 | 4 | 11⅜ | 13½ | V | S | 4 | 10,000 | Nat. | 1,000 |
| | | Pwr. Add. | Gen. | Wor | 6 | 180 | 514 | 6 | 9 | 10½ | V | S | 4 | 10,000 | Nat. | 1,000 |
| | | Pwr. Add. | Gen. | CP | 2 | 188 | 400 | 4 | 11¾ | 13¾ | V | S | 4 | 10,000 | Nat. | 1,000 |
| Utd. Gas Pub. Svce. Co. | Rodessa, La. | Gaso. Ext. | Comp. | CB | 8 | 1,000 | 125 | 4 | 21½ | 36 | H | TT | 4 | ..... | Ref. | 1,500 |
| Sears, Roebuck & Co. | Dallas, Tex | Air. Con. | Gen. | CB | 1 | 600 | 240 | 6 | 16 | 20 | V | S | 4 | 10,000 | Nat. | 1,000 |
| | | | | | 2 | 120 | 514 | 4 | 9 | 10½ | V | S | 4 | 10,000 | Nat. | 1,000 |
| Peoria | Ill. | Mun. Sewage | Blow. | Wor | 1 | 535 | 327 | 6 | 15 | 17½ | V | S | 4 | 9,500 | Slu. | 550 |
| | | | Gen. | Wor | 1 | 300 | 400 | 5 | 12¼ | 14½ | V | S | 4 | 9,500 | Slu. | 550 |
| Sun Oil Co. | Marcus Hook, Pa. | Standby | Pump | Ster | 1 | 500 | 1,200 | 8 | 8 | 9 | V | S | 4 | 10,000 | Ref. | 1,600 |
| City of Livingston | Tex. | Mun. | Gen. | CB | 1 | 450 | 225 | 5 | 18 | 20 | V | S | 4 | 10,000 | .... | .... |
| U. S. Engineers | Tucumcari, N. M. | Govt. | Gen. | CB | 3 | 420 | 400 | 8 | 11½ | 13½ | V | S | 4 | 10,000 | Nat. | 1,000 |
| Old Dutch Ref. Co | Muskegon, Mich. | Ref. | Comp. | CB | 1 | 400 | 200 | 2 | 19 | 20 | H | Tw | 4 | 10,000 | .... | .... |
| Birch Oil Co. | Brea, Calif. | Ref. | Alt. | Cla. | 1 | 300 | 500 | 6 | 9 | 11 | V | S | 2 | ..... | Nat. | 1,000 |
| | | | | | 1 | 250 | 500 | 4 | 9 | 11 | V | ... | 2 | ..... | Nat. | 1,000 |
| Stanolind Pipe Line Co. | Tulsa, Okla | Pipeline | Comp. | ..... | 1 | 300 | ... | 8 | .. | .. | V | S | 4 | 10,000 | Nat. | 1,000 |
| Susque. Pipe Line Co. | McConnellsburg, Pa. | Pipeline | Pump | Ster. | 1 | 300 | 800 | 8 | 8 | 9 | V | S | 4 | ..... | Nat. | 1,000 |
| Toledo | Ohio | Mun. Sewage | Pump | ..... | 1 | 300 | 400 | .. | .. | .. | V | S | .. | ..... | Slu. | 600 |
| | | | Gen. | ..... | 1 | 225 | 514 | .. | .. | .. | V | S | .. | ..... | Slu. | 600 |
| Utd. Gas Pub. Svce. Co. | Houston, Tex. | Aux. Pwr. | Gen. | CB | 1 | 255 | ... | 4 | 14¾ | 16 | V | S | 4 | 10,000 | Nat. | 1,000 |
| | | | Comp. | CB | 1 | 85 | ... | .. | .. | .. | H | ... | 4 | 10,000 | Nat. | 1,000 |
| Phillips Pet. Co. | Vinita Park, Okla. | Booster | Comp. | Cla. | 30 | 250 | ... | .. | .. | .. | H | S | 2 | 10,000 | Nat. | 1,000 |
| Wilson-Snyder Mfg. Co. | Braddock, Pa. | Ref. | Pump | CB | 1 | 235 | 200 | 2 | 17 | 20 | H | Tw | 2 | 10,000 | Ref. | 1,500 |
| | | Ref. | Pump | CB | 3 | 155 | 250 | 2 | 13½ | 18 | H | Tw | 2 | 10,000 | Ref. | 1,500 |
| E. H. Moore & Sons | Tulsa, Okla. | Oil Fld. | Comp. | CB | 3 | 230 | 200 | 2 | 18 | 20 | H | Tw | 4 | 10,000 | Nat. | 1,000 |
| E. H. Jennings | Pittsburgh, Pa. | ..... | Comp. | CB | 1 | 230 | 200 | 2 | 18 | 20 | H | Tw | 4 | 10,000 | Nat. | 1,000 |
| Carter Oil Co. | Tulsa, Okla. | Ref. | Comp. | CB | 2 | 230 | 200 | 2 | 18 | 20 | H | Tw | 4 | 10,000 | .... | .... |
| Barnsdall Oil Co. | Tulsa, Okla. | Ref. | Comp. | CB | 4 | 230 | 200 | 2 | 18 | 20 | H | Tw | 4 | 10,000 | .... | .... |
| Sabine Valley Gaso. Co., Inc. | Shreveport, La. | Ref. | Comp. | CB | 1 | 230 | 200 | 2 | 17 | 20 | H | Tw | 2 | 10,000 | .... | .... |
| Madison Met. Sew. Dist. | Madison, Wis | Sew. Disp. | Blow. | CB | 1 | 230 | 514 | 8 | 9 | 10½ | V | S | 4 | ..... | Slu. | 600 |
| Cedar Rapids | Iowa | Sew. Disp. | Gen. | Wor | 1 | 210 | 300 | 5 | 12½ | 14½ | V | S | 4 | 10,000 | Slu. | 600 |
| Sanit. Dist. | Los Angeles, Cal. | Mun. Sew. | Pump | Cla. | 1 | 200 | 550 | 4 | 9 | 11 | H | S | 4 | 12,000 | Slu. | 600 |
| Enid Ice & Fuel Co. | Enid, Okla. | Pwr. | Comp. | CB | 1 | 190 | ... | 3 | 14¾ | 16 | V | S | 4 | 10,000 | Nat. | 1,000 |
| | | | Gen. | | 1 | 135 | ... | 3 | 11½ | 14 | V | S | 4 | | | |
| Springfield | Ill. | Sew. Add. | Blow. | Cla. | 1 | 177 | 514 | 6 | 9 | 11 | V | S | 2 | 12,000 | Slu. | 600 |
| Campbell, Wyant & Gannon Fdry. | Muskegon, Mich. | Indus. | Comp. | CB | 1 | 160 | ... | 3 | 14¾ | 16 | V | S | 4 | .... | .... | .... |
| Port Lavaca | Tex. | Mun. Ref. | Comp. | Atl. | 2 | 155 | 512 | 6 | 8 | 10½ | V | S | 4 | 12,000 | Nat. | 1,000 |
| Pure Oil Co. | Chicago, Ill. | Oil field | Bandwheel | CB | 1 | 150 | 180 | 2 | 14½ | 20 | H | Tw | 2 | 10,000 | Nat. | 1,000 |
| | | | Pump | CB | 2 | 50 | | | | | H | ... | 4 | 10,000 | Nat. | 1,000 |
| City of Durham | N. C. | Sew. Disp. | Gen. | CB | 2 | 150 | 360 | 4 | 10½ | 13½ | V | S | 4 | 11,000 | Slu. | 750 |
| Atlantic Pipe Line Co. | Philadelphia, Pa. | Pipeline | Pump | CB | 1 | 150 | 327 | 2 | 12½ | 15 | H | Tw | 2 | 10,000 | Nat. | 1,000 |
| Ann Arbor | Mich. | Mun. Sew. | Blow. | ..... | 1 | 120 | ... | .. | .. | .. | V | S | .. | ..... | Slu. | 600 |
| Healey Petroleum Co. | Bradford, Pa. | Ref. | Pump | CB | 1 | 120 | 240 | 2 | 12½ | 15 | H | Tw | 2 | 10,000 | .... | .... |
| Gilliland Genl. Gaso. Co. | Gladewater, Tex. | Ref. | Comp. | CB | 2 | 115 | 200 | 1 | 17 | 20 | H | S | 2 | 10,000 | .... | .... |
| Geo. L. Reed Mllg. Co. | Brookville, Pa. | Indus. | Gen. | CB | 1 | 110 | 220 | 2 | 12½ | 15 | H | Tw | 2 | 10,000 | .... | .... |
| Oil Well Supply Co. | Dallas, Tex. | ..... | Pump | CB | 1 | 100 | 300 | 2 | 11 | 15 | H | Tw | 2 | 10,000 | .... | .... |
| | | | | | 1 | 45 | 450 | 2 | 9 | 10 | V | S | 2 | 10,000 | .... | .... |
| Sullivan Machry. Co. | Dallas, Tex. | ..... | Comp. | CB | 1 | 70 | 300 | 1 | 12½ | 16 | H | S | 2 | 10,000 | .... | .... |
| Bi-Rite Food Mkt. | Los Angeles, Cal. | Store Pwr. | Gen. | Wauk. | 1 | 67 | 1,000 | 6 | 5 | 5½ | V | S | 4 | 11,200 | Nat. | 1,120 |
| | | | | Wauk. | 1 | 36 | 1,000 | .. | 4½ | 5¾ | V | S | 4 | 11,200 | Nat. | 1,120 |
| Palo Alto | Calif. | Mun. Sew. | Alt. | Pitt. | 1 | 60 | 900 | 4 | 6 | 8 | V | S | 4 | ..... | Slu. | 657 |
| Ontario | Calif. | Mun. Sew. | Blow. | Ent. | 1 | 45 | 600 | 4 | 6½ | 8½ | V | S | 4 | 14,500 | Slu. | 630 |
| Stanolind Oil & Gas Co. | Tulsa, Okla. | Pipeline | Pump | CB | 3 | 40 | 250 | 1 | 12½ | 16 | H | S | 2 | 10,000 | Nat. | 1,000 |
| Magnolia Petroleum Co. | Dallas, Tex. | Pipeline | Pump | CB | 3 | 40 | 250 | 1 | 12½ | 16 | H | S | 2 | 10,000 | Nat. | 1,000 |
| Niagara Oil Corp. | Bradford, Pa. | Pipeline | Pump | CB | 12 | 30 | 180 | 1 | 11 | 15 | H | S | 2 | 10,000 | Nat. | 1,000 |

**Abbreviations**—Gen.—Generator, Alt.—Alternator, Comp.—Compressor, Blow.—Blower, CB—Cooper-Bessemer Corp., Wor—Worthington Pump & Machry. Corp., IR—Ingersoll-Rand Co., RS—Rathbun-Jones (subsidiary of Ingersoll-Rand), Ster.—Sterling Engine Co., CP—Chicago Pneumatic Tool Co., Cla.—Clark Bros. Co., Atl.—Atlas-Imperial Diesel Engine Co., Ent.—Enterprise Engine Co., Pitt.—Pittsburg Motor Co., Wauk.—Waukesha Motor Co., TT—Twin-Tandem, Tw—Twin, S—Single, Nat.—Natural gas, Slu.—Sludge or sewage gas, Ref.—Refinery Gas.




steady fuel flow. One German engine even runs on hydrogen by using a special mixing valve and injecting water.

Variations in B.t.u. content are usually met by changing mixing-valve proportions. Compression pressure should be 110 to 115 lb. for natural gas and refinery gas, 70 to 80 lb. for pure butane or free hydrogen above 40 to 45 per cent (because of lower ignition point, to avoid pre-ignition), 140 lb. for producer gas, and 160 lb. for blast-furnace gas. In any case, compression should be kept as high as possible for efficiency.

In general, fuel heat goes a third to power, a third to jacket water and a third to exhaust. A gas engine of any size will provide about ⅔ lb. of steam per b.-hp.-hr., providing the engine is heavily loaded continuously and the waste-heat boiler is close to the exhaust valve. Tube maintenance in such boilers is usually high, although Bethlehem Steel Co. has recently installed three vertical single-pass boilers on three of its 5,000-hp. engines and reports quite satisfactory results. Total heat recovery is 70% without taking the gas below its dewpoint.

# I. C. FUELS

EARLY diesels were heavy-duty, slow-speed units with large cylinders and air-blast injection of fuel. They literally would burn almost any liquid with B.t.u. content from whipping cream to tar. Modern engines, under stress of competition both from other prime movers and from other designs, obtain much higher economy (one manufacturer has cut fuel consumption, in lb. per b.hp.-hr. from 0.60 with semi-diesel in 1920 to 0.46 in 1924 and 0.38 today with diesel, an increase in efficiency from 22.3% to 35.2%). Most engines use mechanical injection of fuel, and avoid air blowers, fuel centrifuges and preheaters.

Fuel oil is either a byproduct obtained in crude-oil refining, or crude oil itself. Regardless of source, chemical composition is about the same: 83-87% carbon 11-14% hydrogen, and oxygen, nitrogen and sulphur in amounts from traces to 2 or 3%. Carbon and hydrogen possess the highest heats of combustion of any two elements, 14,500 and 52,000 B.t.u. per lb., but neither is suitable alone as a fuel, except in highly specialized engines (the Erren engine will burn pure hydrogen, the Rupamotor pulverized anthracite, which is almost pure carbon). The combination of the solid and the gas, however, produces a readily combustible mixture in air.

But what is a good diesel fuel? In the old days, engines differed so much that a fuel that was meat for one engine was poison for all the others. Manufacturers asked for oil samples and tested them, but found that two batches of reputedly the same oil would vary widely in characteristics. Sometimes oil vendors mixed two oils in stock to get the gravity and viscosity specified—with all sorts of effects on the other properties. The A.S.M.E. and the S.A.E. appointed a joint committee which did much to clarify and orient the situation. But there is still much to be done. A recent survey of manufacturers by Hubner and Murphy *(National Petroleum News, Aug. 14, 1935)* gave these average requirements:

| | Operating Speed, R.P.M. | | |
|---|---|---|---|
| | Above 1,000 | 500-1,000 | Below 500 |
| Viscosity, SU @ 100 deg. F. | 36–78 | 41–75 | 42–100 |
| A.P.I. Gravity | 26–35 | 26–38 | 24–34 |
| Sulphur, % max. | 0.2 | 1 | 1.2 |
| Hard Asphalt, % max. | 0.53 | 0.3 | 0.5 |
| Conradson carbon, % max. | 0.5 | 1 | 2 |
| Ash, % max. | 0.08 | 0.1 | 0.12 |
| Water & Sediment, % max. | 0.5 | 0.6 | 0.9 |
| Flash point, deg. F. min. | 152 | 140 | 150 |
| Pour point, deg. F. max. | 10-15 deg. below oprtg. temp. | | |
| Heat value, B.t.u./lb., min. | 18,960 | 18,850 | 18,750 |

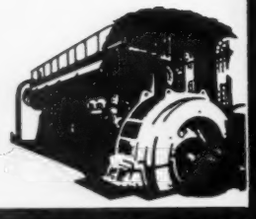
The three most important fuel qualities, in order, were: ignition quality, cleanliness and viscosity; yet some manufacturers felt that others were more important.

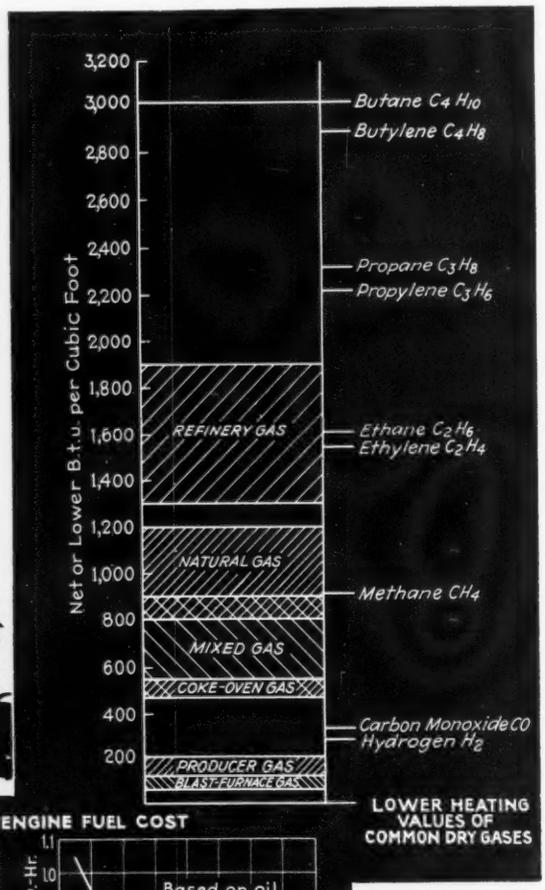
Most refiners now supply standard diesel fuels that will operate the stationary diesel without difficulty. In each case, however, the engine builder should be questioned regarding the suitability of a particular fuel.

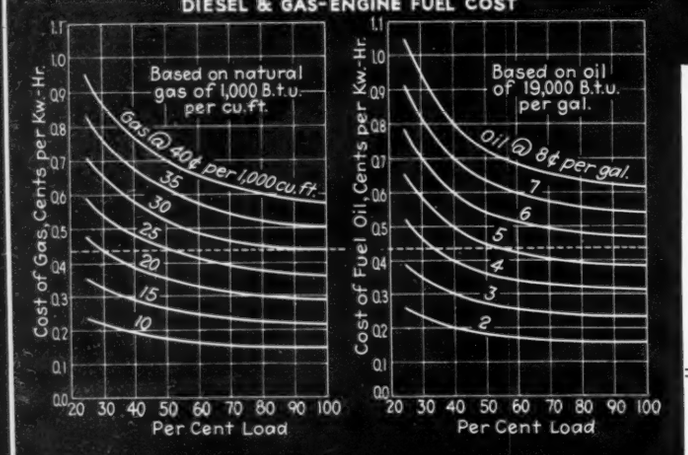
Generally speaking, the gas engine is not so finicky about its fuel as is the diesel, mainly because its fuel is a gas, readily mixed with air. While a specific modern diesel is limited in fuel, a given modern gas engine is likely to be able to burn a wide range of fuels with slight changes in the mixing valve.

Fuel for gas engines vary in B.t.u. content from a blast-furnace gas at 90 to 150 B.t.u. per cu.ft. to butane at 3,000 B.t.u. and over. Natural gas, the commonest fuel, is principally methane, with some ethane and the next higher hydrocarbon, and possibly some oxygen, nitrogen, hydrogen and carbon monoxide. Its heating value is about the same as that of pure methane, about 900 B.t.u. per cu.ft. (lower or dry value). The range may be 800 (when nitrogen, $CO_2$, etc., are high) to 1,200 (when there is much ethane in the mixture). About 1 cu.ft. of air is required to burn 1 cu.ft. of blast-furnace gas—or roughly a cubic foot for every 100 B.t.u. in the gas. Thus natural gas requires 10 cu.ft. of air per cu.ft. of gas, and butane requires 32 cu.ft. The gas must be clean and free from corrosive elements, a constant danger particularly with refinery gases, which almost always have some hydrogen sulphide content. If $H_2S$ runs above 0.25% by volume, the gas must be cleaned. Usually a simple scrubber will do it, in one case reducing $H_2S$ from 200 grains per cu.ft. (0.33%) to 5, cost averaging about a cent per 1,000 cu.ft. Since refinery gas runs about 1,500 B.t.u. per cu.ft., a cent per 1,500,000 B.t.u. is cheap insurance against engine failure.

Gas as a fuel has these advantages: It is clean. Usually no investment in storage facilities is required to use it. It is easy to handle. Natural-gas rate structures are usually much more lenient on demand charges than are equivalent electric rates. Price is almost always stable, and the gas is not paid for until a month after use.






Comparative heating values of various gases, and, left, an approximate chart for fuel costs on diesel and gas engines. A rough comparison of fuel costs can be made, showing whether or not a detailed study is advisable

# GOVERNING

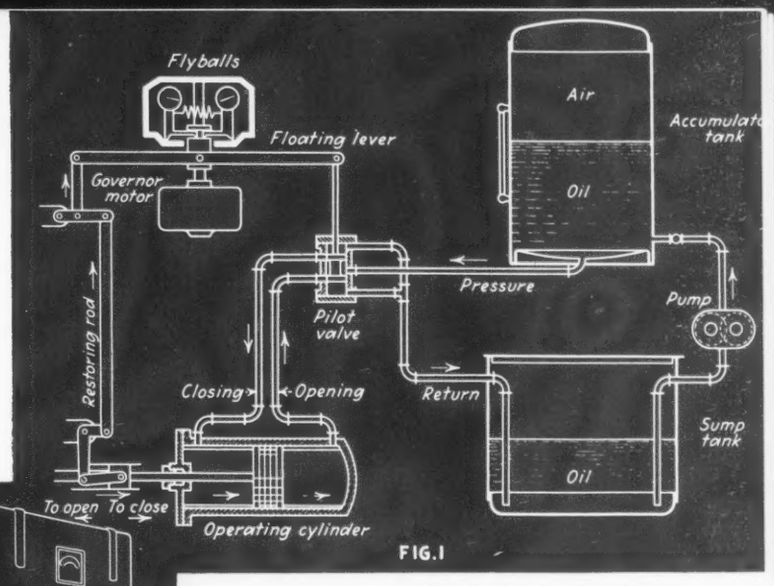


FIG. 1

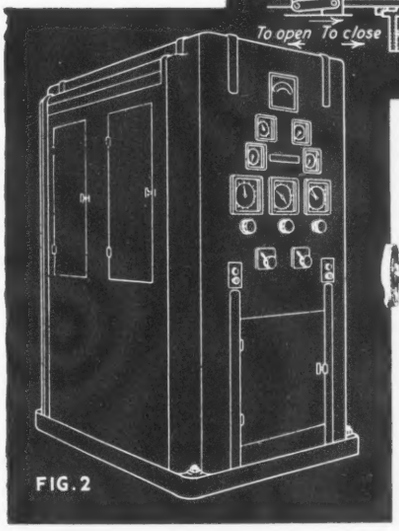


FIG. 2

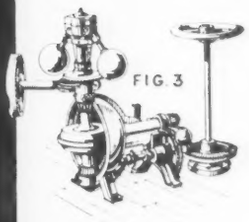


FIG. 3

**Fig. 1—A hydraulic-turbine governing system in its simplest form. Fig. 2—A modern hydraulic governor is a far cry from its prototype of 30 years ago, Fig. 3. On the modern governor all equipment is inclosed in a cabinet, with only the indicators, gages and control knobs outside**

*By S. Logan Kerr*
*Mechanical Engineer*

TO SAY that "a prime mover is no better than its Governor," is putting it mildly. An engine or turbine out of control races and is likely to cause a catastrophe; unless a prime mover's speed can be regulated within close limits it is not suited for modern power requirements. Consequently, along with prime-mover development, governors have been improved until they now do things thought impossible a few years ago.

Four major types of prime movers—steam engines, steam turbines, hydraulic turbines and diesel engines (much of this discussion also applies to gas engines)—are used for electric power generation. For each of these the governor will have certain elements in common. Other features will not be required in all types and may be found in only one.

Almost all types of speed-governing equipment include a centrifugal element, sensitive to changes in speed, and sometimes an inertia element which responds to rate of speed change. The usual governor head consists of two flyballs, the force of which is opposed by a spring mechanism which increases its resistance in proportion to the centrifugal force acting upon the flyballs. This provides a stable mechanism which has a definite position for each speed. If centrifugal force of the flyballs increases faster than the resisting force of the springs, the governor becomes unstable.

Construction details of centrifugal elements vary widely with different types of governors. In general, the flyballs are small and light, and rotate at a speed between 400 r.p.m. and 1,000 r.p.m., depending upon prime mover and design of governor. Heavy slow-speed governor flyballs on early designs were pivoted in ordinary bearings. Later ball or roller bearings were used, and for extremely light high-speed types, hardened knife edges. More recently, ball bearings have been used widely.

Opposing springs may be of the single or double type. Two springs are frequently used in parallel, one on either side of the governor head. In some cases, the two springs have their ends connected to both flyballs, and in others each spring acts between a flyball and a fixed support. With the latter construction, careful adjustment is necessary to equalize the springs and their settings, while, in the first arrangement the springs automatically adjust for slight differences in characteristics.

Early types of steam-engine governors used a dead weight to resist the action of centrifugal forces, and stability was secured by a linkage which reduced the lifting effect of the flyball element as the speed increased. The steam-engine control mechanism was operated directly by the flyballs. It adjusted the throttle valve or varied the cutoff to regulate steam input to the engine.

Those early engine units usually operated independently, no special provisions being necessary to provide for parallel operation. Modern prime movers driving electric generators made necessary improvements in governing mechanisms, particularly with a.c. generators, to permit parallel operation without hunting or interchange of power between units.

For steam turbines, power-input adjustment is accomplished by varying the quantity or pressure of the steam. On small turbines, the throttle valve may be operated directly by the governor. Auxiliary power may be required if the throttle is large. Relays or power amplifying means are then used, and the governor actuates a small pilot valve which in turn controls movement of the main throttle.

In some types, the number of nozzles admitting steam to the turbine blades is varied, and the control mechanism is moved by a piston in a steam or oil cylinder, controlled by the governor pilot valve. For large units, multiple throttles and nozzle controls are required to secure a smooth flow of power, to reduce the size of the stroke. For a more detailed description of steam-engine and turbine governor mechanisms, see pages 305 and 316.

Control mechanisms for varying the flow through a hydraulic-turbine runner are large and invariably operated by some power-amplifying means through the action of the governors. An oil-pressure system, or occasionally a water-pressure system, is used in the servomotors. For some early units, the power for operating the turbine inlet mechanism came from the main unit through governor-controlled clutches. A modern hydraulic-turbine governing system in its simplest form is shown schematically in Fig. 1. The pump takes oil from the sump tank and stores it under pressure in the accumulator tank to operate the servomotor that controls turbine-gate position. The governor flyballs are motor driven with the floating lever pivoted to a collar on the governor shaft. The right-hand end of the floating lever is connected to the pilot valve and the left-hand end to the restoring rod.

Assume load is dropped off the turbine. This will produce a rise in speed, which will cause the flyballs to move out to a new position. In doing so they will lift the right-hand end of the floating lever, and the pilot valve opens ports to admit pressure oil to the left-hand end and discharge oil from the right-hand end of the servomotor cylinder. This causes the servomotor to start closing the gates. As the gates move toward the closed position, the restoring-rod mechanism raises the left-hand end of the floating lever and causes its right-hand end to close the pilot-valve and stop gate movement when balance has been restored between turbine power input and output. Modern governors have several refinements, but the diagram shows only basic elements.

Inlet mechanisms to waterwheels gen-

erally are multiple wicket gates, but sometimes are cylinder gates. A recent design is a single wicket-type gate at the scroll-case entrance. Runner blades of Kaplan wheels are usually adjusted also with the wicket gates. In some European installations, wicket gates have been omitted, and governing is done by adjusting the runner blades. A more recent form developed in the United States has a balanced runner blade that adjusts its pitch automatically as the wicket gates are moved.

On impulse turbines, single or multiple needle valves direct the jet on the buckets. The long water column in the penstock prevents any rapid change in flow, therefore various auxiliary means, such as a bypass nozzle, Fig. 14, a deflector, Fig. 12, pressure regulators, Fig. 13. Other devices cut off the jet from the wheel to permit more rapid change in power input than could be obtained by adjusting the needle alone.

Diesel engines have many different types of control, but all vary the fuel injected into the individual cylinders. In some cases a common rail operates the individual fuel-injection valves. Others have fuel pumps that operate continuously with a bypass or cutoff to vary the fuel to the injection system. In other engines the fuel is injected by air, and air and fuel is controlled in combination.

Present tendency in governor design for all four types of prime movers is toward relay governors. On smaller units, self-contained power amplifying equipment such as oil pumps and small actuating pistons are included as integral parts of the governor.

## Speed Droop

With the advent of a.c. generators and parallel operation of a number of prime movers of different types and characteristics, it became necessary to provide a controllable speed-droop element in the governor. This device causes a change in speed from no-load to full-load and is designated as inherent speed drop, speed droop, no-load full-load regulation, and other terms. "Speed droop" is expressed as a percentage of normal speed, Fig. 4.

The amount of speed droop required for different prime movers and different operating conditions varies from zero to 6% or more. In a small isolated plant having a stable load, where the governor is equipped with adequate anti-racing devices, it is possible to set the speed droop to zero and have the same speed at all points of operation.

Where two or more generators are operating in parallel and where the load fluctuates widely and rapidly, it may be necessary to set speed droop to a value as high as 6% or more. When load fluctuations do not exceed the capacity of a single unit (preferably not more than a fraction of its capacity) the governor on one unit can be set for zero speed droop. Governors on the other units are then set for some value of load that will permit them to carry a steady load, while the burden of regulation falls to the single unit with no speed droop. This plan has been carried out successfully in a number of small isolated municipal diesel-electric generating stations.

Where power-system size and load fluctuation increase beyond the capacity of one unit, it is necessary to set all governors with a reasonable amount of speed droop and permit the base speed to vary slightly or provide automatic means to correct frequency following a load change.

Speed droop is obtained in a number of ways. On early governors for steam turbines or diesels, an auxiliary spring was added which altered the characteristics of the flyball slightly with respect to load. Action of this spring is permanent where the spring or dashpot action in the anti-racing device has a temporary action only. Hydraulic and relay-type governors in general use a cam mechanism or a link and lever mechanism to secure speed-droop characteristics.

Protecting prime movers against excessive speed presents a serious problem. There are many instances where the gov-

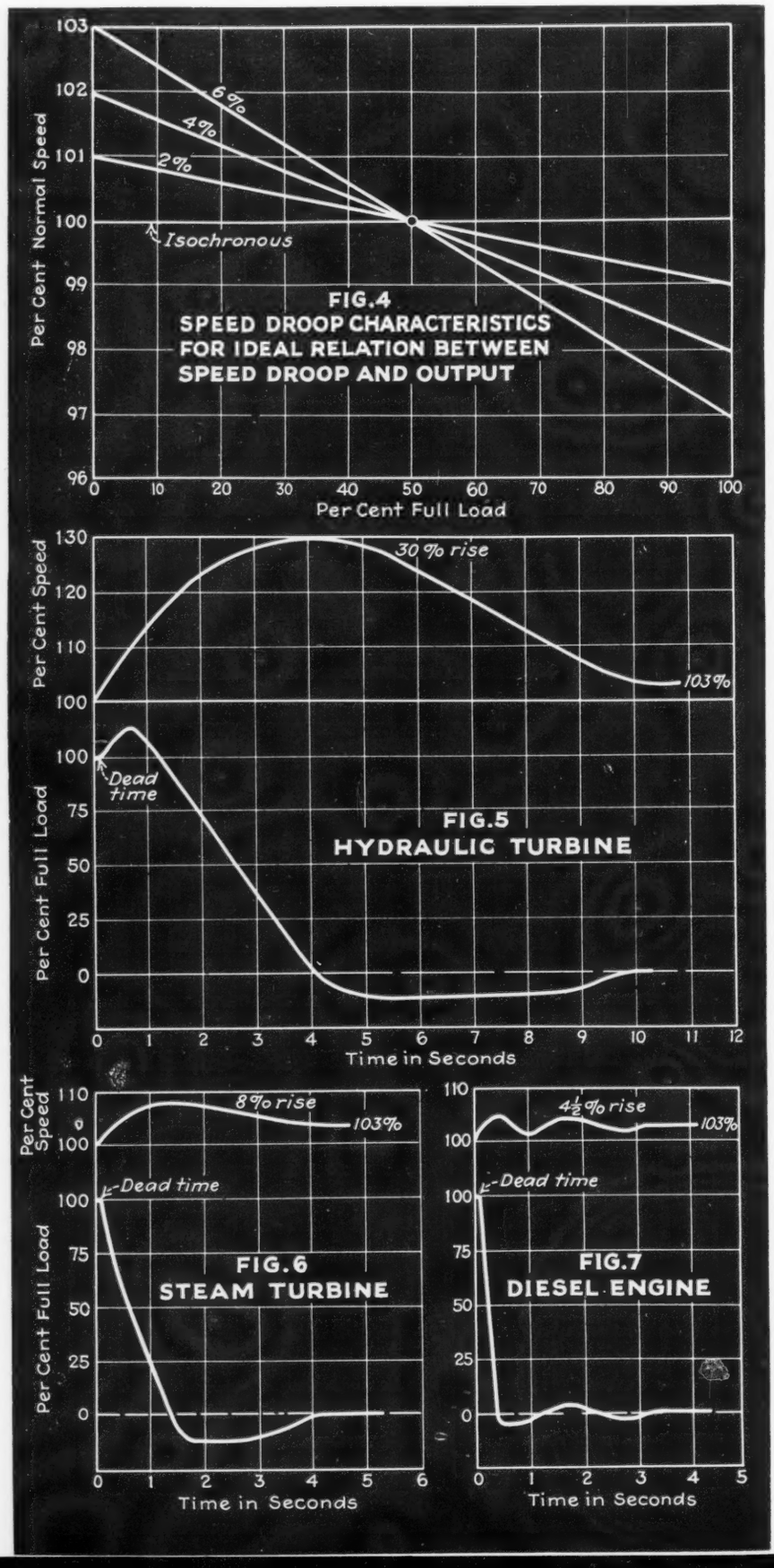



Fig. 4—Speed-droop characteristics for ideal relation betwen speed droop and power output. Figs. 5 to 7—Curves showing speed change resulting from loss of full load from a waterwheel, a steam turbine and a diesel engine, respectively

ernor has failed on steam engines and the unit speeded up sufficiently to burst the flywheel. With diesel engines the same danger exists, while with steam turbine normal speeds can not be exceeded by any wide margin without seriously endangering the safety of the machine. Hydro-electric units have a definite maximum speed. Tests on model runners show that the overspeed can rarely exceed twice normal, unless the head fluctuates widely, in which case maximum speed may reach three times normal. Most all hydro-electric generators and turbines are designed to withstand safely a speed from 80 to 100% above normal. Special overspeed protection is applied only in special instances, such as in automatic plants without an operator. On diesel and steam engines and steam turbines overspeed trip devices are standard equipment and are usually installed as auxiliary devices separated entirely from the governor so that failure of the governor will not impair overspeed-trip operation.

Friction is one of the most troublesome factors affecting governor operation. This reduces flyball, pilot valve or control response, decreases sensitivity, and is the most frequent source of governor troubles.

Inertia of the governor, main-control-mechanism and main-rotor all cause a time lag between change of load demand and beginning of governor-corrective action. Assuming a sudden decrease of load, the speed must decrease, flyballs move, pilot valve open, and control mechanism be actuated before the change in power input begins. This total time varies from 1/50 sec. in extremely accurate governors up to ¼ sec. or even more in large governors having heavy control gates to operate. This time lag or dead time (See Figs. 5 to 7) increases the duration and rate of momentary speed changes and permits a wider departure of speed from normal.

In the power medium itself, certain problems arise which affect governor behavior greatly. With a diesel engine, these are reduced to a minimum since the fuel supply can be cut off almost instantaneously or can be increased very rapidly. Only a small volume of fuel is required, it has little inertia, and except for compressibility of the fuel and expansion of fuel conduits between pump and cylinder, the pump causes immediate response.

With steam turbines of multi-stage types and with compound connections a large quantity of steam is in the turbine beyond the throttle valve. This steam must expand down to a low pressure before power input to the turbine is reduced to the speed no-load value. This time varies from 1/10 sec. up to a full second, depending upon the size, type and number of stages of the turbine. Operating time of the throttle valve can be made very rapid in the case of the steam turbine, but the large valves, auxiliary oil or steam cylinders, required to operate to make it close, limit the traversing time of the governor from full opening to closed position to usually not less than 0.25 sec. It may be a full second.

## Hydraulic Turbines

For hydraulic turbines, the traversing time is limited to some extent by gate-mechanism size, but this could be a period of one second or less if mechanical operation were the limiting factor. The column of water contained in the passages leading to the turbine and in the draft tube has so much inertia that it must be accelerated or retarded at a gradual rate. In small open-flume units where water velocities are slow and water passages short it is frequently possible to reduce the traversing time of the gate mechanism to 1.5 or 2.0 sec. As the water velocities and length of water passages increase, traversing time must be greatly increased. It is rarely possible or desirable to move the turbine gates from full-open to full-closed in less than 3 or 4 sec. For the largest units, it may be necessary to increase this time to 5 sec. to avoid

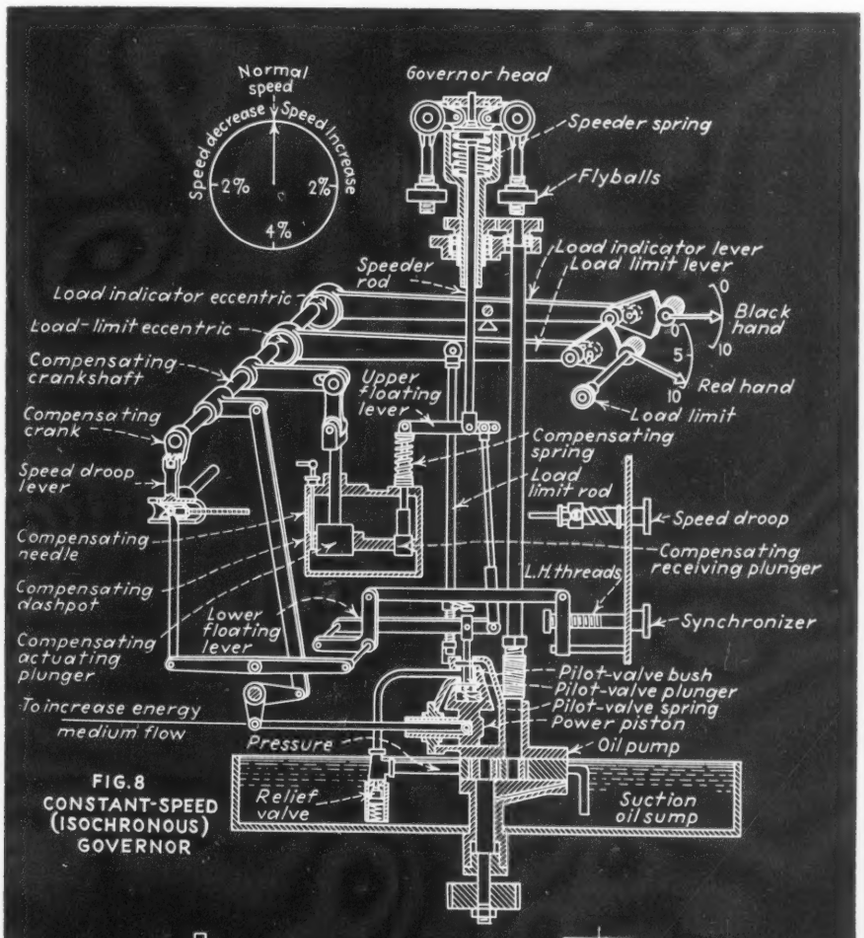



FIG. 8 CONSTANT-SPEED (ISOCHRONOUS) GOVERNOR

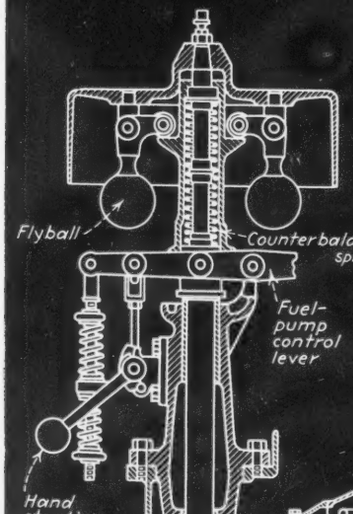



FIG. 9 FLYBALL GOVERNOR

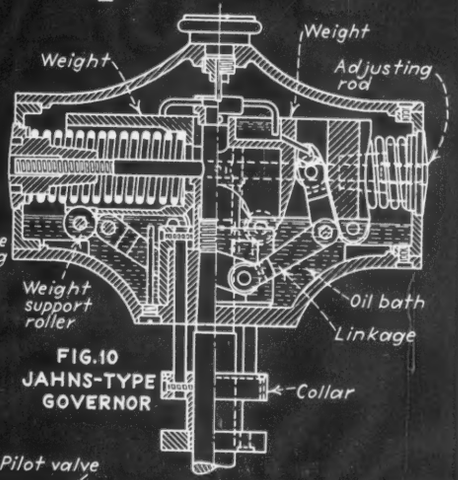



FIG. 10 JAHNS-TYPE GOVERNOR

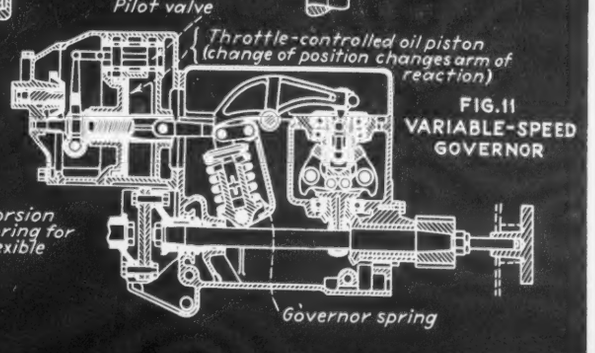



FIG. 11 VARIABLE-SPEED GOVERNOR

Fig. 8—Elements of an isochronous diesel governor. Fig. 9—A flyball governor for a diesel. The flyballs act through bell cranks on the sleeve, which in turn acts on a lever and linkage to the fuel pumps. The hand lever at left is for starting and stopping. Fig. 10—Jahns-type diesel governor, a sensitive type utilizing centrifugal force to move weights. Fig. 11—A variable-speed diesel governor, in which throttle position affects action

breaking the water column of the draft-tube.

With extremely long pipelines, many auxiliary devices are employed to assist in regulating the water column and permit the governor to regulate speed with reasonable satisfaction. Surge tanks are frequently installed to allow a substantial reduction in governor time. In other cases, particularly with impulse wheels or high-head reaction turbines, it is customary to employ bypass valves or relief valves, Fig. 14, which are open as the gates are closed and bypass the water rather than changing its velocity suddenly. A slow-closing feature is usually provided on such valves to reduce pressure rise resulting from operation and to prevent waste of water which would otherwise be continually discharged through them. A modification of this scheme is used frequently in high-head impulse wheels. A jet deflector, Fig. 12, deflects, the jet from the buckets, until the needle nozzle is gradually closed at a safe rate and the deflector returned to normal position. A relief valve or deflector does not assist in taking care of oncoming loads. If wide fluctuations in demand will occur on a plant having a long pipeline, it is necessary to provide a surge tank or to permit waste of a certain amount of water by having the deflector partially enter the jet or by having the bypass or relief valve stay open.

Inertia of the control mechanism and of the power medium has already been discussed. It is necessary in all prime-movers to supply some flywheel effect to bridge the gap between instant of load change and time at which power supply equals the new demand. In hydro-electric units the flywheel effect must be great, as time delay is long. The high speed at which steam turbines operate provides a large amount of kinetic energy which assists in bridging this gap between power supply and demand. In diesel engines the time delay is very small and the flywheel effect is provided to smooth out the intermittent action of the individual cylinders and deliver a uniform steady output.

The character of load supplied by the prime mover has a decided effect upon governor performance. A steady load will permit a uniform speed without demand being placed on the governor. With fluctuating loads, the governor is called upon to act more frequently and to correct for changes in demand. When load fluctuations occur at regular intervals, as in certain industrial applications, the governor may be required to act continuously and give the appearance of hunting. In such cases the governor may be blocked temporarily while the operator sees whether fluctuations continue unchanged or increase. Governor action will usually reduce the speed fluctuations in such cases. Occasional instances have been found where the period of load fluctuation corresponds with the normal period of governor operation and the speed fluctuations were augmented by the governor. The remedy was to adjust the rate of governor response by speeding up its action or by increasing effectiveness of the restoring mechanism.

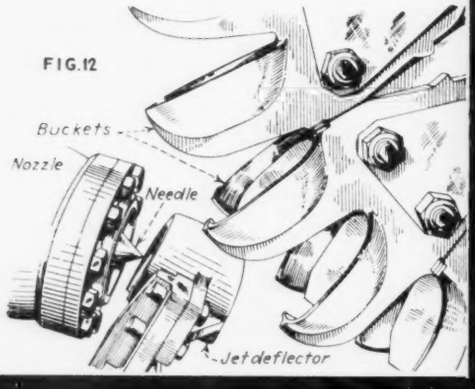
Interconnected high-tension transmission systems have emphasized the necessity for precise speed control of prime movers. Governor sensitivity used to be on the order of 1%, that is, if the speed departed 1% from normal the governor would act. Maximum momentary departure of speed from normal was usually many times this amount. Guarantees are now being called for which require movement of the governor valve for a speed change of 0.01 of 1%. The true sensitivity may be several times this amount. Over-all sensitivity is the speed change required to effect a change in prime-mover input and includes all of the inertia, friction and lost motion between the governor and device controlling the flow of steam, water or fuel to the prime mover.

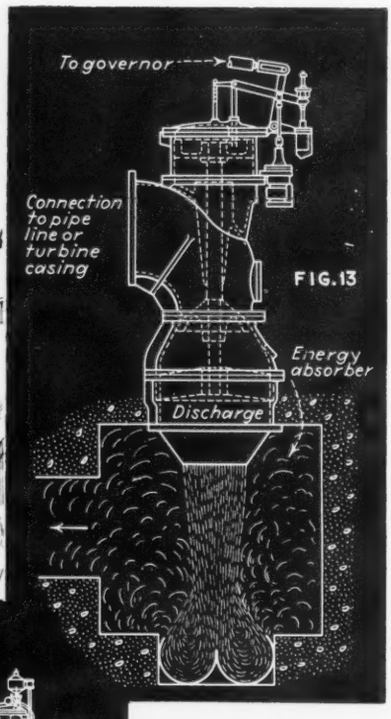
Maximum momentary speed changes for hydraulic-turbine units are fixed by governor-traversing time, flywheel effect of the rotating element and inertia effect of the water column. Dead time or degree of sensitivity of the governor is important but will effect the maximum speed rise only to a very small degree.

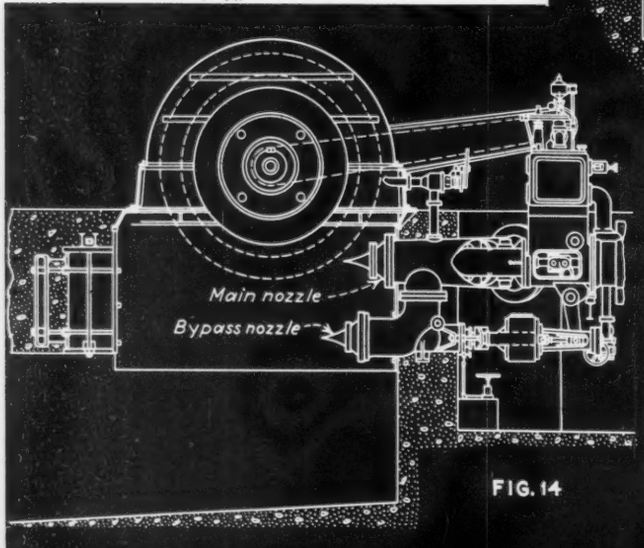
With steam turbines, the rotating energy, amount of steam trapped in the turbine beyond the throttle valves, as well as traversing time of the throttle valve, will control the maximum momentary speed rise. This will rarely exceed 10% of normal speed and will usually not be more than 5 or 6% for full-load tripped off the turbine, Fig. 6, compared with 25% to 50% rise in speed with a normal hydraulic turbine, Fig. 5. In some cases the hydraulic unit can be held to a speed rise not in excess of 8 and 10% where jet deflectors are employed with impulse units.

The diesel, however, offers best conditions for speed control, since the fuel supply is acted upon directly and time delays are extremely small. Diesels driving a.c. generators have been tested, and the maximum speed rise for full load thrown off has been less than 4% of normal speed and about equal to the amount of speed droop as set by the governor adjustment, Fig. 7.

The most recent change in governors is in their appearance. Early governors were covered with a multitude of attachments; the flyball element was open and exposed to dirt and rust, lever and line mechanisms were cumbersome, and the whole arrangement was frequently unsightly. From time to time, improvements were made—the flyballs were housed in, the control mechanisms were placed in a base casting and attachments arranged neatly inside a cover. Now all governing equipment, pumps and sump tanks for hydraulic turbines have been placed in a cabinet with only the necessary control handles showing and with the dials for meters and indicators on a panel forming part of the cabinet, as in Fig. 2. Smaller governors have also undergone major changes in appearance, with all parts enclosed, and dials and control knobs on the outside (See Fig. 2, page 317).



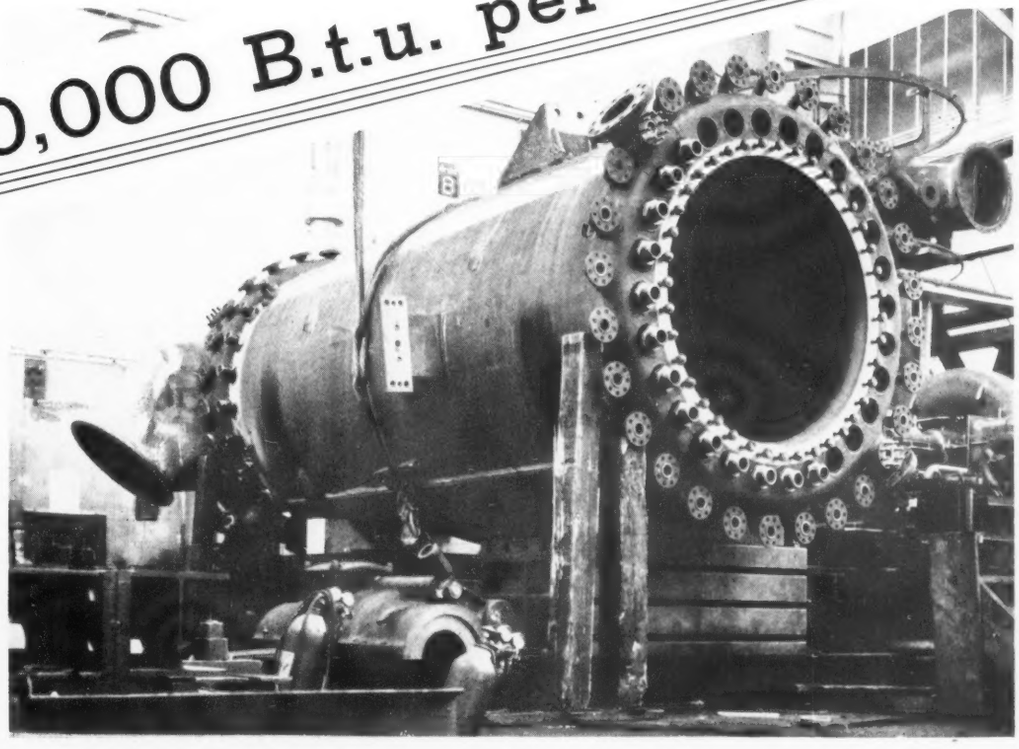



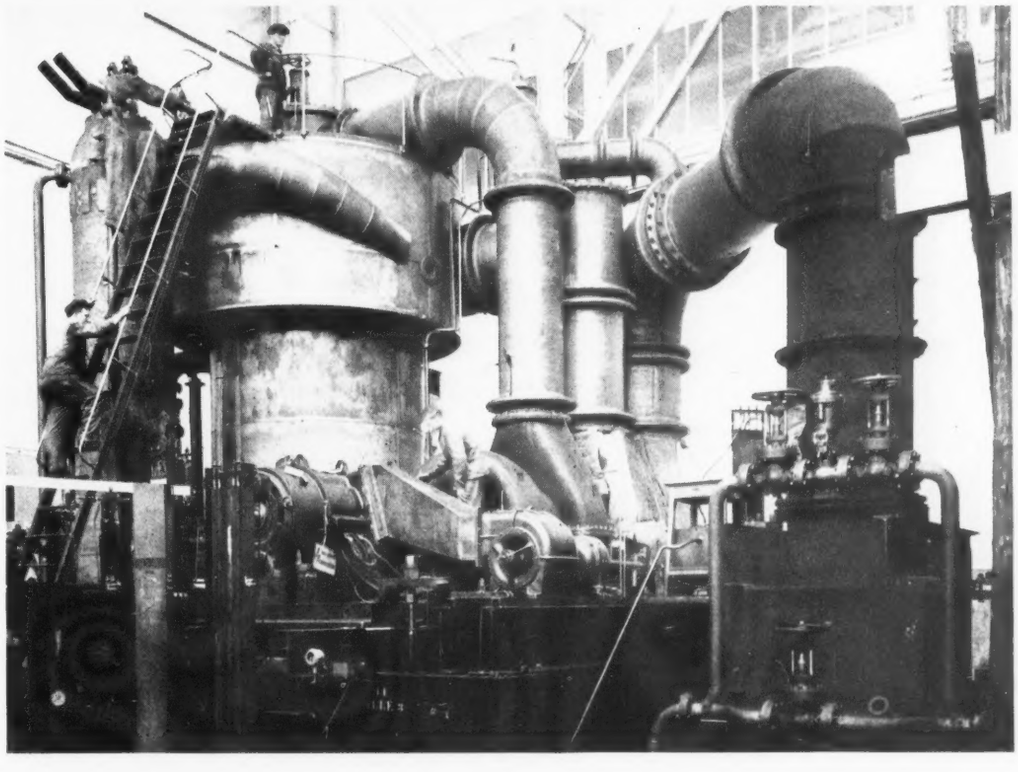




Figs. 12 to 14—Three methods of controlling flow to an impulse wheel Fig. 12—A jet deflection. Fig. 13—Pressure regulator. Fig. 14—By-pass nozzle

AMERICA'S first Velox boiler, a 50,000-lb. per hr. unit, will soon be installed in an Eastern plant. Illustrated is one of two 165,000-lb. per hr. units for Oslo, Norway. Features include pressure combustion, gas turbine run by flue gas, forced water circulation, furnace heat release about 1,000,000 B.t.u. per cu. ft.

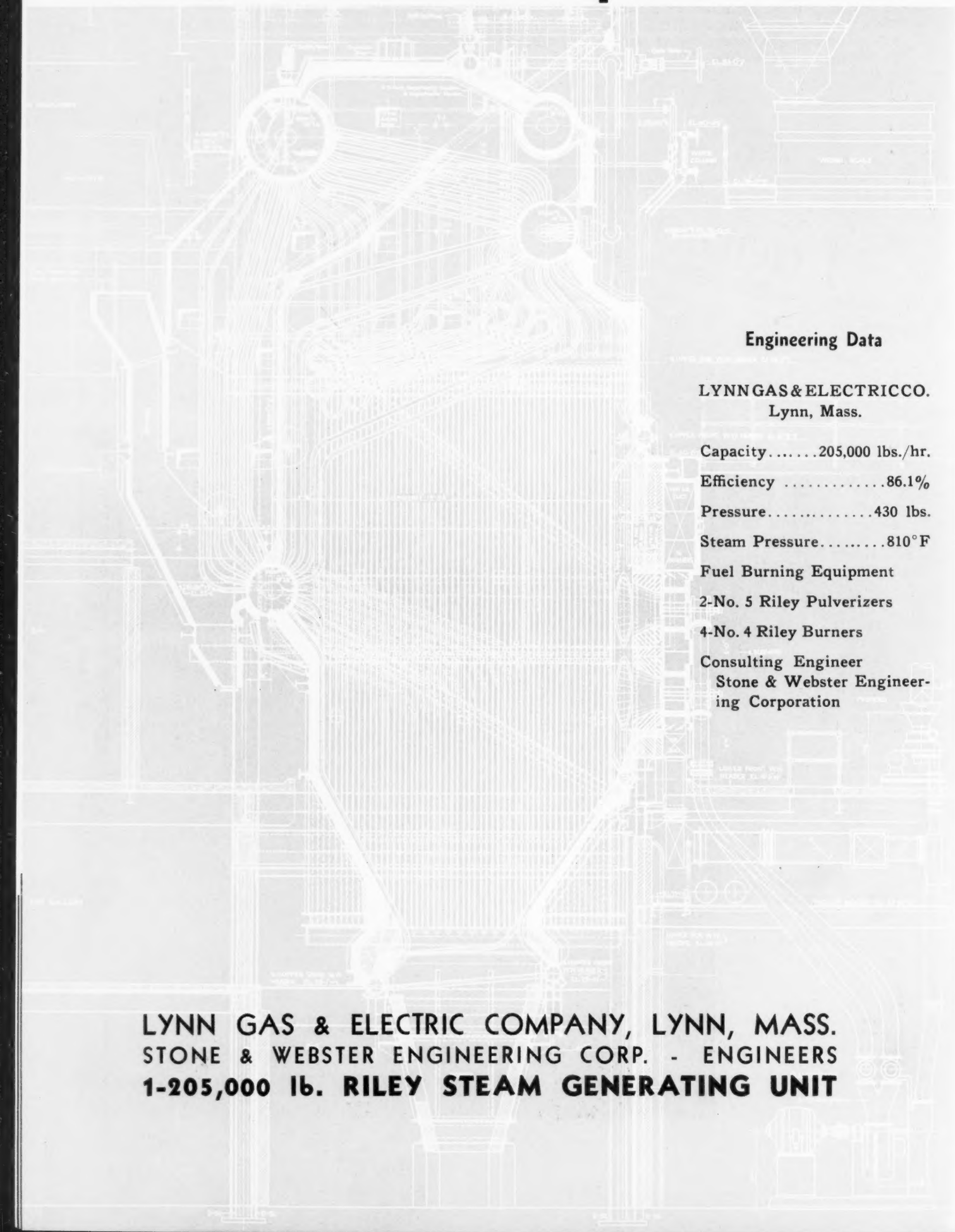
# 1,000,000 B.t.u. per Cu. Ft.





Top—Steam generator lying on its side. Note center combustion space and ring of vertical, multiple-walled, forced-circulation water tubes. Bottom — Left, steam separator. Next Velox generator. Foreground, gas turbine driving turbo-compressor for combustion air (18 lb. gage).

# ...for their most recent plant extension



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Capacity.....205,000 lbs./hr.

Efficiency .....86.1%

Pressure.....430 lbs.

Steam Pressure.....810°F

Fuel Burning Equipment

2-No. 5 Riley Pulverizers

4-No. 4 Riley Burners

Consulting Engineer
Stone & Webster Engineering Corporation

LYNN GAS & ELECTRIC COMPANY, LYNN, MASS.
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**1-205,000 lb. RILEY STEAM GENERATING UNIT**

# LYNN GAS & ELECTRIC COMPANY

## *selects a* RILEY

## STEAM GENERATING UNIT

For the most recent addition to their boiler plant, the Lynn Gas & Electric Company, through their engineers, the Stone & Webster Engineering Corporation, selected Riley Steam Generating Unit . . . following the general swing to Riley.

It is quite significant that so many companies should select Riley Steam Generating Units when they modernize or increase the capacity of their plants. In many cases these companies have had experience with many other types of steam generating equipment and would not have swung to Riley unless they had been definitely impressed by the performance and appearance of existing recent Riley installations.

Engineers everywhere are talking about the outstanding performance, complete satisfaction, the absolute reliability, the excellent engineering design and physical appearance of Riley Steam Generating Units. That is why there has been such a decided swing to Riley Steam Generating Units. Why plant after plant has adopted Riley Units in preference to all others.

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1—160,000 lbs./hr. 500 lbs.-750°F
Gas Fired
Valco, Inc., Engrs.
Arthur T. Nelson, Cons. Engr.

Lynn Gas & Electric Co.
Lynn, Mass.
1—205,000 lbs./hr. 430 lbs. - 810°F
Pulverized Coal Fired
Stone & Webster Engineering Corp.
Engineers

Holyoke Water Power Co.
Holyoke, Mass.
1—200,000 lbs./hr. 675 lbs. - 750°F
Oil and Pulverized Coal Fired
Arthur T. Nelson, Engineer

Standard Oil Co. of California
Richmond, California
3—125,000 lbs./hr. 850 lbs. - 760°F
Stone & Webster Engineering Corp.
Engineers

*West Virginia Pulp & Paper Co.
Williamsburg, Pa.
2—100,000 lbs./hr. 600 lbs. - 675°F

Socony-Vacuum Oil Co.
for Naples, Italy
2—60,000 lbs./hr. 750 lbs. - 700°F

Imperial Paper & Color Co.
Glens Falls, N. Y.
2—60,000 lbs./hr. 300 lbs.

Tsishuyen Electricity Works, China
1—110,000 lbs./hr. 233 lbs. - 680°F

*Large Chemical Plant
2—100,000 lbs./hr. 600 lbs. - 650°F

University of Minnesota
Minneapolis, Minn.
1—100,000 lbs./hr. 300 lbs.

Upper Michigan Power & Light Co.
Escanaba, Mich.
1—90,000 lbs./hr. 410 lbs. - 650°F
W. I. Barrows, Engineer

*Repeat orders.

# Weight of Copper, Brass & Steel Wire, Rods & Bars

| Gage No. | Brown & Sharpe | | | Washburn & Moen | | Diam. Across Flat Inches | Hexagon | | | Square | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | Diam. Inch. | Weight per 100 ft. | | Diam. Inch. | Wt. per 100 ft. | | Weight per Foot | | | Weight per Foot | | |
| | | Copper | Brass | | Steel | | Copper | Brass | Steel | Copper | Brass | Steel |
| 000000 | .580 | 101.93 | 96.88 | .4615 | 56.87 | 1/8 | .052 | .050 | .046 | .060 | .057 | .053 |
| 00000 | .517 | 80.99 | 76.98 | .4305 | 49.48 | 5/32 | .081 | .078 | .072 | .093 | .088 | .082 |
| 0000 | .460 | 64.11 | 60.94 | .3938 | 41.41 | 3/16 | .117 | .112 | .104 | .136 | .129 | .120 |
| 000 | .410 | 50.93 | 48.41 | .3625 | 35.09 | 7/32 | .161 | .153 | .142 | .184 | .176 | .163 |
| 00 | .365 | 40.37 | 38.37 | .3310 | 29.25 | 1/4 | .208 | .199 | .184 | .240 | .229 | .212 |
| 0 | .325 | 32.00 | 30.42 | .3065 | 25.08 | 9/32 | .263 | .252 | .233 | .303 | .289 | .268 |
| 1 | .289 | 25.31 | 24.05 | .2830 | 21.38 | 5/16 | .326 | .311 | .288 | .375 | .358 | .332 |
| 2 | .258 | 20.17 | 19.17 | .2625 | 18.40 | 11/32 | .394 | .377 | .348 | .454 | .433 | .401 |
| 3 | .229 | 15.89 | 15.10 | .2437 | 15.86 | 3/8 | .467 | .446 | .414 | .540 | .515 | .478 |
| 4 | .204 | 12.61 | 11.99 | .2253 | 13.55 | 13/32 | .551 | .526 | .488 | .636 | .607 | .563 |
| 5 | .182 | 10.04 | 9.54 | .2070 | 11.44 | 7/16 | .636 | .608 | .564 | .735 | .703 | .651 |
| 6 | .162 | 7.95 | 7.56 | .1920 | 9.84 | 15/32 | .731 | .698 | .647 | .843 | .805 | .746 |
| 7 | .144 | 6.28 | 5.97 | .1770 | 8.36 | 1/2 | .833 | .795 | .737 | .960 | .917 | .850 |
| 8 | .128 | 4.96 | 4.72 | .1620 | 7.01 | 17/32 | .940 | .899 | .833 | 1.085 | 1.037 | .960 |
| 9 | .114 | 3.94 | 3.74 | .1483 | 5.87 | 9/16 | 1.054 | 1.008 | .933 | 1.217 | 1.160 | 1.077 |
| 10 | .102 | 3.15 | 3.00 | .1350 | 4.87 | 19/32 | 1.175 | 1.100 | 1.040 | 1.353 | 1.293 | 1.199 |
| 11 | .091 | 2.51 | 2.38 | .1205 | 3.88 | 5/8 | 1.303 | 1.245 | 1.153 | 1.500 | 1.434 | 1.328 |
| 12 | .081 | 1.99 | 1.89 | .1055 | 2.97 | 21/32 | 1.437 | 1.371 | 1.271 | 1.654 | 1.582 | 1.465 |
| 13 | .072 | 1.57 | 1.49 | .0915 | 2.24 | 11/16 | 1.576 | 1.505 | 1.395 | 1.816 | 1.737 | 1.608 |
| 14 | .064 | 1.24 | 1.18 | .0800 | 1.71 | 23/32 | 1.720 | 1.645 | 1.524 | 1.982 | 1.898 | 1.757 |
| 15 | .057 | .984 | .936 | .0720 | 1.38 | 3/4 | 1.875 | 1.790 | 1.660 | 2.163 | 2.067 | 1.914 |
| 16 | .051 | .788 | .749 | .0625 | 1.04 | 25/32 | 2.034 | 1.942 | 1.800 | 2.341 | 2.241 | 2.075 |
| 17 | .045 | .614 | .583 | .0540 | .779 | 13/16 | 2.200 | 2.104 | 1.948 | 2.536 | 2.426 | 2.246 |
| 18 | .040 | .485 | .461 | .0475 | .602 | 27/32 | 2.372 | 2.262 | 2.099 | 2.736 | 2.614 | 2.420 |
| 19 | .036 | .393 | .373 | .0410 | .449 | 7/8 | 2.545 | 2.436 | 2.256 | 2.942 | 2.811 | 2.603 |
| 20 | .032 | .310 | .295 | .0348 | .323 | 29/32 | 2.739 | 2.615 | 2.421 | 3.145 | 3.014 | 2.791 |
| 21 | .028 | .238 | .226 | .03175 | .269 | 15/16 | 2.920 | 2.802 | 2.594 | 3.380 | 3.229 | 2.990 |
| 22 | .025 | .189 | .180 | .0286 | .218 | 31/32 | 3.128 | 2.991 | 2.769 | 3.600 | 3.445 | 3.190 |
| 23 | .023 | .160 | .152 | .0258 | .178 | 1 | 3.333 | 3.186 | 2.950 | 3.840 | 3.672 | 3.400 |
| 24 | .020 | .121 | .115 | .0230 | .141 | 1 1/16 | 3.762 | 3.595 | 3.329 | 4.338 | 4.146 | 3.839 |
| 25 | .0179 | .097 | .092 | .0204 | .111 | 1 1/8 | 4.217 | 4.031 | 3.732 | 4.865 | 4.649 | 4.305 |
| 26 | .0159 | .077 | .073 | .0181 | .0874 | 1 3/16 | 4.708 | 4.499 | 4.166 | 5.422 | 5.182 | 4.798 |
| 27 | .0142 | .061 | .058 | .0173 | .0799 | 1 1/4 | 5.209 | 4.979 | 4.610 | 6.006 | 5.740 | 5.315 |
| 28 | .0126 | .048 | .046 | .0162 | .0701 | 1 5/16 | 5.740 | 5.486 | 5.080 | 6.620 | 6.327 | 5.858 |
| 29 | .0113 | .0387 | .0368 | .0150 | .0601 | 1 3/8 | 6.304 | 6.025 | 5.579 | 7.266 | 6.944 | 6.430 |
| 30 | .0100 | .0303 | .0288 | .0140 | .0523 | 1 7/16 | 6.887 | 6.583 | 6.095 | 7.941 | 7.589 | 7.027 |
| 31 | .0089 | .0240 | .0228 | .0132 | .0465 | 1 1/2 | 7.614 | 7.277 | 6.738 | 8.646 | 8.263 | 7.651 |
| 32 | .0080 | .0194 | .0184 | .0128 | .0437 | 1 5/8 | 8.797 | 8.408 | 7.785 | 10.15 | 9.697 | 8.979 |
| 33 | .0071 | .0153 | .0145 | .0118 | .0372 | 1 3/4 | 10.22 | 9.766 | 9.043 | 11.76 | 11.24 | 10.41 |
| 34 | .0063 | .0120 | .0114 | .0104 | .0289 | 1 7/8 | 11.71 | 11.19 | 10.36 | 13.52 | 12.92 | 11.96 |
| 35 | .0056 | .0095 | .0090 | .0095 | .0241 | 2 | 13.35 | 12.75 | 11.81 | 15.38 | 14.70 | 13.61 |
| 36 | .0050 | .0076 | .0072 | .0090 | .0216 | 3 | 29.97 | 28.65 | 26.52 | 34.61 | 33.08 | 30.63 |

Approximate weight of one cubic foot of steel 490 lbs., copper 554 lbs. and brass 533 lbs.



Courtesy The Louis Allis Co.

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# PRACTICAL AIDS TO OPERATION

## Test Condensate for Oil Content

OIL in steam boilers is a frequent cause of overheating, bulging, and even explosions. Any trace of oil should be removed from boilers, and kept out of all feedwater and condensate returns. Various sources of trouble of this kind should be kept in mind. These include all reciprocating steam machinery exhausting steam for condensate returns. Indirect oil heaters in which even a pin hole in the steam coil can cause trouble. Any other indirect or direct contact between oil and steam are potential sources of trouble.

Where such equipment is installed, frequent checks should be made for oil in the condensate. While a chemical analysis is required for an accurate determination of the exact amount of oil, a more practical test will serve operating requirements. Visible indication, such as oil showing in the boiler gage glass, usually comes too late to prevent trouble

A ¼-in. valved tap off of the feedwater line from the boiler-feed pump provides an easily applied test. Place a piece of white cotton cloth over this tap and discharge feedwater through it for several minutes. Any trace of oil or other foreign matter, if present, will show on the cloth. It is unlikely that oil will emulsify at ordinary feedwater temperatures to such a degree that it cannot be seen by such a test. This test should be made periodically, and immediate corrective steps taken if oil is discovered.

*Boston, Mass.* C. O. DIBBLE

## Dry Pipe Explodes Inside Steam Drum

HOW MANY engineers have heard of a dry pipe in a boiler exploding? This one exploded inside of the boiler drum while the pressure was 410 lb. per sq.in. The typical type of internal collecting pipe was used, Figs. 1 and 2. The latter also shows the superheater location. This unusual accident occurred twice before investigation determined the cause. Each time the internal collecting pipe was found opened up. The boiler had been in standby service just previous to each explosion. When the explosions occurred, steam pressure jumped from 410 to 450 lb., and dropped back to about 410. This pressure fluctuation took place just after lighting the fire, and it happened practically instantaneously.

As the boiler was fired by pulverized coal, it was possible to cause an intense heat when forcing the fires. This condition existed particularly when starting the boiler from standby service in an emergency. Standby pressure was 400 lb. and line pressure was 650 lb. When an emergency called for this boiler, the fires were lighted and forced.

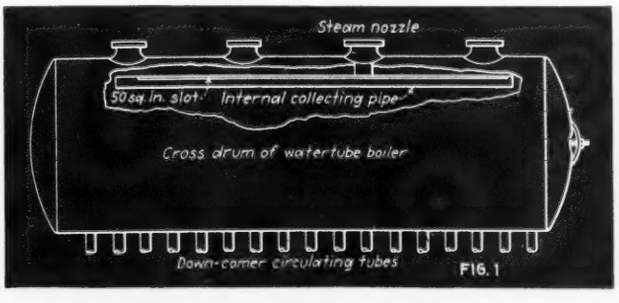


FIG. 1

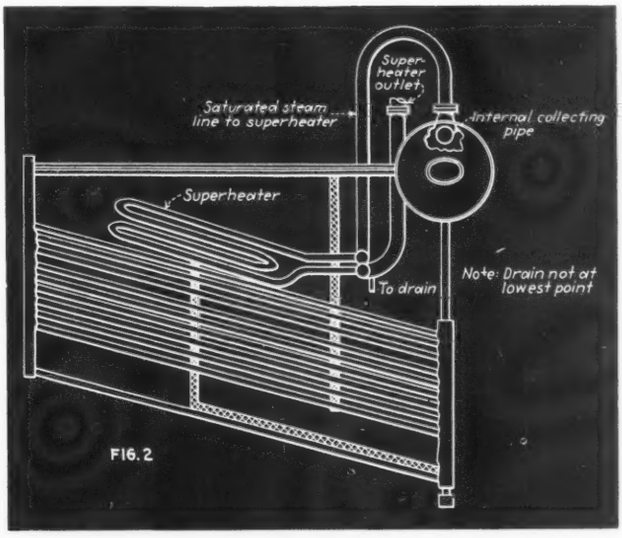


FIG. 2

Cross-sectional area of piping between boiler steam nozzle from the internal collecting pipe to superheater is 78.54 sq.in. The collecting-pipe opening area is a little over 50 sq.in. After the boiler had stood idle for some time, condensate collected in the superheater. When the burners were lighted and forced, this condensate flashed into steam. The stop valve at the superheater outlet being closed, the sudden accumulation of steam pressure backed up into the internal collecting pipe. Since the aggregate area of the collecting-pipe openings is about 36.5% less than the cross-sectional area of the steam pipe, the former offered a restriction to steam flow, and a sudden rise of pressure occurred inside the dry pipe and blew it apart. This pipe is usually of light gage bolted construction, as in this installation, not designed to withstand a considerable internal pressure.

*Cleveland, O.* A. C. WIGGINS

## May Appear Impossible But True Nevertheless

TOO MANY installations of new power plants have been made, only to show inexcusable later errors in piping or general mechanical layout. Three such installations will illustrate the point.

A rather amusing case occurred in a laundry boiler plant where two horizontal tubular boilers were installed. Installation of auxiliary equipment and piping was made by a local contractor and steam fitter from the blueprints of the designer after the boilers were erected. When installation was completed, a boiler inspector was called to make an inspection before starting up. He found the steam lines from the boilers turned back from an elbow and nipple on the front steam nozzle. The steam piping was directly above the upper manhole openings and so close to them that it was impossible for a man to enter the boilers. An extra day's work was necessary, plus the additional expense of new piping to relocate the steam mains.

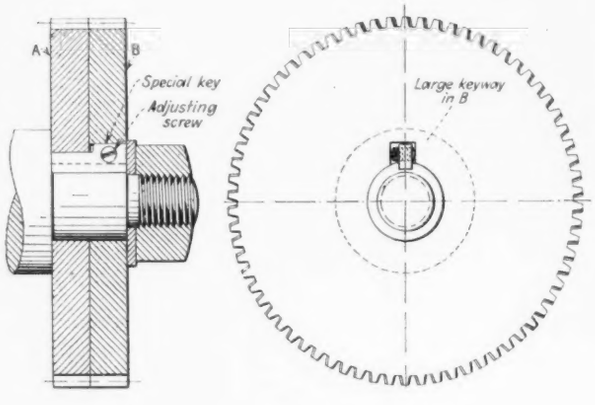
Two heating boilers were recently installed in a mercantile

building. The operating engineer noticed that neither stop nor check valves were provided in the condensate return piping to the boilers. He pointed this out to one of the designers during installation, but no attention was paid to him. Three months after the plant was in operation, the fireman started up one of the boilers and neglected to open the stop valve. When several pounds pressure accumulated in the steam space of the boiler, the water was forced out into the return system and the boiler badly overheated. Needless to say, the necessary stop and check valves were installed in the return piping after the expense of repairing the damaged boiler.

A 5,000-sq.ft. water-tube boiler fired by oil was installed in a paper mill. Design of the boiler setting neglected to consider explosion doors. While the operating personnel brought this to the attention of the proper parties, it was decided that the explosion doors were unnecessary. Installation of two explosion doors would have cost about $40 additional. In less than a year after this plant started operation, miosture in the oil caused a serious flare-back and furnace explosion. Part of one side wall was blown down and the baffles were badly damaged. Repairs to baffles and boiler setting cost $125. Fortunately no one was injured, but such an accident might easily have caused more serious results.

A chief engineer who is competent to take charge of power-plant operation ought to have had extensive training in operation under a wide variety of conditions. Such a man should be consulted during the time of design for major changes or new installations in the plant where he is to have charge. His experience should be valuable towards making suggestions in operating conditions and methods encountered only in years of operating experience.

*Chicago, Ill.* R. O. BILLINGS




## Adjustable-Keyway to Take Backlash From Gears

Two gears on a magnetic brass-dust separator had a rather short life due to the material handled. A small amount of backlash interfered with proper working of the machine. When the gears were renewed for the second time I decided to make them so that the backlash could be removed.

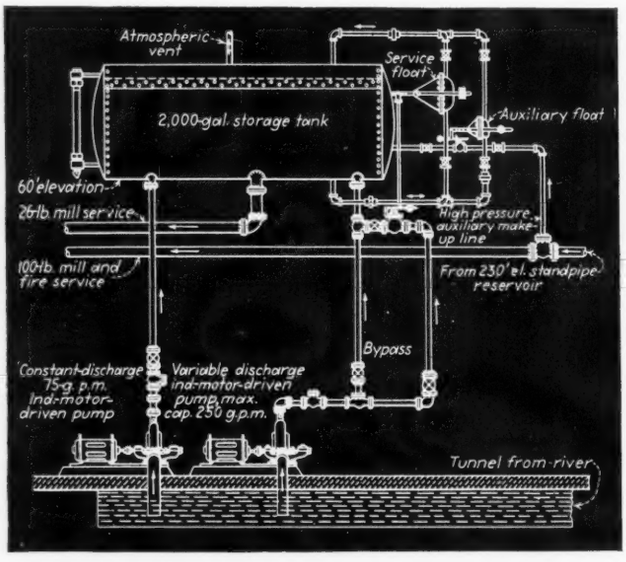
One gear was made solid as before, but the other was built in halves, as shown in the diagram. Turning up two blanks, I clamped them together and cut the teeth as if they were one piece. In one half, *A*, an ordinary keyway was cut to match the one in the shaft and in the other half, *B*, a deeper and wider keyway was made. A special key was made and tapped for a round-end screw, this screw being exactly equal in length to the width of the keyway in *B*. When tooth wear produces backlash, gear *B* is removed and the screw adjusted to throw the teeth forward a little until the teeth in both halves fit snugly in the other gear. Turning the screw does not affect its fit in the keyway, so that the gear can be moved in either direction without slipping.

*Hamilton, Ont.* H. MOORE

## Tags Valve Stems For Safety

RISING-STEM valves, particularly if installed in a horizontal position, may create a safety hazard. When the valve is open, the stem may protrude a foot or more. If the valve is near a passageway that is danger of someone running into the stem and being injured. To prevent this in our plant I made up round tin disks about 4 in. in diameter and hung one on the end of each horizontal valve stem, by means of wire. It hangs down like a label and always has a slight motion, due to air currents, which easily attracts attention.

*Passaic, N. J.* MARK BELL




## Pump System for Additional Water Supply

A NEW ENGLAND paper mill recently expanded and installed additional paper-making machinery. It was found at this time that the installed standpipe did not have sufficient capacity to supply the increased maximum demand for water. As this standpipe was still in good condition, its replacement was thought unwarranted. Several plans for additional water supply from a nearby river were studied.

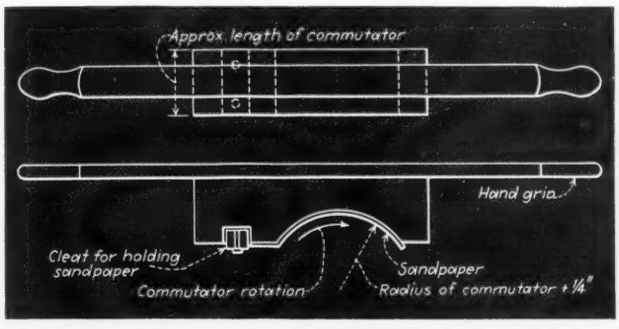
The problem was solved by separating the low- and high-pressure water piping into two systems. A 2,000-gal. storage tank was installed at the highest part of the mill, 60 ft. above the machine floor level, sufficient elevation for all low-pressure purposes.

A constant-speed, low-head, 75-g.p.m. centrifugal pump was connected to the river-water tunnel to supply the average minimum demand to the storage tank. A second pump having a maximum discharge capacity equal to the difference between the average minimum and average maximum demand (250 g.p.m.) was installed. A study of local conditions showed that the varying demand would cause too frequent

starting and stopping of the second pump if its motor drive was controlled by a float switch. A float-operated valve was therefore installed to control the discharge of this pump, as in the diagram. The 75-g.p.m. pump operated at constant discharge, all the time when the plant was operating, to supply just over the minimum demand. The variable discharge 250- g.p.m. pump operated at fluctuating rating to supply the peak demands.

An auxiliary float-operated valve controlled an emergency supply line from the standpipe so that failure of a pump or an abnormal peak in demand would not affect continuous supply of water to the low-pressure system.

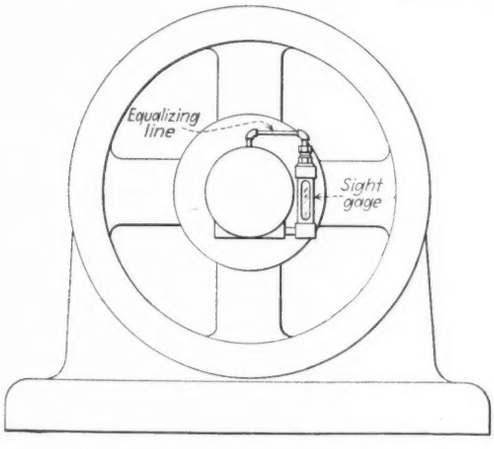
*Chicago, Ill.* CHAS. A. ARMSTRONG




## Device for Truing Commutators

THE diagram shows a simple device that I use for keeping the surface of commutators in good condition. It is made of a wood block attached to a suitable handle to facilitate using. The block is squared so that its width is about equal to commutator length. A curved surface is cut toward one end of the block and at the other end is cut a slot for a cleat to hold several strips of sandpaper. The curve should be cut to a radius equal to that of the commutator plus ¼ in. This will give ample space for several pieces of sandpaper and have the paper make good contact with the commutator. One end of several strips of sandpaper are attached to the block by the cleat. When one strip wears out, it is torn off and a new one exposed.

*St. Marys, O.* J. L. YOUNG




## Equalizing Air Pressures Cures Bearing Trouble

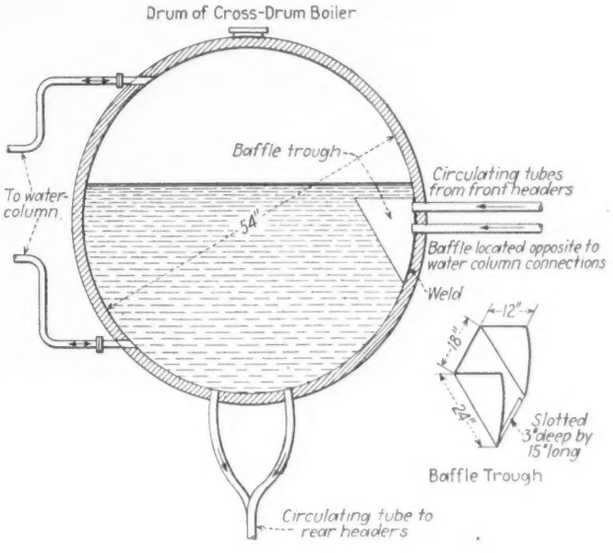
SUCTION created by ventilating motor windings often causes oil to creep along the rotor shaft. With large motors, requiring great quantities of cooling air, reduced air pressure on the motor side of the bearings may result in false indications on bearing-reservoir oil-level indicators. This condition was discovered recently on a 300-hp. motor driving a boiler-feed pump in an industrial power plant. Any attempt to maintain normal oil level as indicated by the sight gage caused oil to flow into the motor windings.

This condition was remedied by equalizing air pressures inside and outside of the bearing housing. The bearing housing was made air-tight at all points, with the exception of the shaft opening. To equalize bearing air pressures and vent the oil-level indicating gage, an equalizing line was installed, as in the diagram. This change eliminated oil seepage and resulted in correct indication of oil lever.

*Richmond, Va.* C. R. ANDERSON

## Overheated Boiler Caused By Water Column

A CROSS-DRUM water-tube boiler operating at 1,200-lb. pressure overheated causing distortion and leakage of upper rows of tubes. Indications were that the trouble was caused by low water. Feedwater regulators were tested and found to be operating satisfactorily. Boiler operators were very conscientious and attentive to their duties, and it was thought that they were not at fault. Numerous other points were considered, but nothing could be found that could have caused the trouble.




It was finally discovered that when forcing the boiler, water flow through the horizontal circulating tubes was so great that it carried across the boiler drum into the lower water-column connection. This action caused the water level indication to be very much higher than the actual water level in the boiler. As the feedwater regulator operated on the water column level, it did not prevent water in the boiler from going too low.

A trough was fabricated from heavy-gage sheet steel as in the diagram. This was spot welded on to the inside of the boiler drum in front of the horizontal circulating tubes and directly opposite the water-column connections. Now operation at high ratings does not cause any irregularity in water-column operation.

*Columbus, O.* J. R. WILLIAMS

# CURRENT COMMENT

## Disadvantages of Curved Bridge Walls

THE argument about the merits and demerits of curved and flat bridge walls in *Power*, Feb., 1936, was interesting. To get at the answer let us consider the purpose of a bridge wall and determine if a bridge wall is necessary to force the flames up to the boiler.

One of the important purposes of a bridge-wall is to retain the fuel on the grates. The hot gases from the furnace are buoyant and ascend rapidly until they are stopped by the boiler. Therefore a curved bridge wall is not required to force the flame up to the boiler. This is true regardless of fuel burned under return-tubular boilers, usually used in plants which are here considered.

A curved bridge wall is objectionable because ashes collect on its top and impair the draft. A man working in the combustion chamber could not pass over it. If the girth seam happens to be over it and a leak occurs, it could not be caulked without taking down the bridge wall. In many instances the ash and soot must be removed over the bridge wall and out the furnace door. This cannot be easily done if the wall is curved. Finally, a curved bridge wall costs more to construct and maintain.

*New York, N. Y.* JAMES W. BLAKE

## Purpose of Bridge Wall

IN THE article "Curved versus Flat Bridge Walls" in February *Power*, I think the purpose of the bridge wall is missed entirely.

In the first place, its purpose is not to make the flames hug the shell of the boiler, for since combustion is completed in the combustion chamber, combustible volatiles coming into contact with the relatively cold surfaces of the shell may be cooled below the ignition point, resulting in incomplete combustion. With a very high or curved bridge wall, a decided damper effect is created in the furnace causing flare-backs and excessive furnace temperature. Wood, or refuse, being high-volatile fuels, require incandescent ignition surfaces such as an ignition arch, and roomy gas passages.

The purpose of the bridge wall is to promote better mixing of volatiles and air, to supply a glowing target wall to assist ignition of the gases, and as one fireman so aptly put it, "to keep the fire from falling into the combustion chamber."

Most of the heat absorbed by the shell of the boiler is due to the radiant heat of the fire and furnace, and it is evident that engineers are not so concerned about not getting the shell hot enough, as there seems to be a ready market for girth-seam protectors.

*Milwaukee, Wis.* L. M. LARSON

## Bridge Wall at Rear of Grate

HAVING had several years of experience burning wood fuel in both hogged and slab form I wish to add my comment to the discussion on "Curved Versus Flat Bridge Walls." One not familiar with the typical southern sawmill might easily wonder at the lack of modern equipment and practices in these power plants. With few exceptions, such equipment could not be justified. Ordinarily the excess wood and refuse over that burned under the boilers was a nuisance and had to be disposed of in an incinerator. Recent developments in the paper-making and wood-fiber industries have created a demand for much of this waste material.

From the information given I assume that the boiler under discussion is a return-tubular type, without forced-draft, hand fired with slabs. Due to the high moisture content of this fuel, green slabs or sawdust, the furnace should have ample volume and a larger area of heated brick work to help dry the fuel. A Dutch-oven type furnace is sometimes required where high boiler ratings are necessary.

A bridge wall should always be used at the rear of the grates, and if properly constructed will withstand firing of solid fuel by hand for a normal life. The wall is necessary for many reasons, chief among which are: It prevents fuel from being thrown past the grates toward the rear of the boiler, which would cause late combustion, the flames sometimes reaching the top of the stack. The extra brickwork increases the furnace temperature due to its radiant heat. This helps to dry the fuel, and raise the air and gas temperature to the combustion point.

Long-leaf yellow pine, because of the oils and resinous products it contains, has a greater amount of volatile and a slightly higher heating value than most other woods. For this reason the products of combustion require a more thorough mixing for efficient operation than for other woods. A properly constructed bridge wall produces a better mixture of air and combustible gases by forcing them together as they pass through the restricted opening. Bridge walls are usually made flat, chiefly because this type is more easily constructed and maintained.

*Waynesboro, Va.* J. M. MYERS

## Cut Down Air Supply

IF A small particle of coal is subjected to sufficient heat it will ignite. In a comparatively cool furnace, especially at light loads, this heat is largely obtained from adjacent burning particles in a self-sustaining frame. If too much air is being used in a burner, or if there is not a steady flow of coal particles, the flame will die out because the effect is lost of one burning particle igniting those adjacent. This is because the distance between individual particles is too great. These unburned particles may accumulate on the floor where they will smoulder and finally ignite causing a flareback.

There are several factors affecting flame stability of pulverized coal burners at light loads. Some of them are: 1. Number and size of burners. 2. Type of burner—straight shot or turbulent, forced or natural draft. 3. Furnace design—amount of water-cooled surface and vertical or horizontal firing. 4. Type of system—storage or unit fired. 5. Coal fineness.

A good pulverized-coal burner should have a peak-to-low-load ratio of about 5 to 1. If a greater range than this is desired, the number of burners should be increased and size decreased. Straight-shot burners cannot be built in capacities as large as a turbulent type and, therefore, a larger number are needed for a given installation. Stable combustion can be obtained by cutting some of the burners out of service for light loads. If the installation is such that burners cannot be taken out of service, the quantity of primary and especially secondary air should be reduced to a minimum.

In a storage system it is a simple matter to reduce the number of burners in service, but in the unit system we often find one pulverizer and one burner on small jobs and on the larger jobs combinations of two or three pulverizers with two or three burners for each pulverizer. Of these combinations, the toughest to operate at light loads is the one pulverizer and one burner installation. The best way to handle this at light loads is to use just barely enough primary air to keep the pulverizer from choking and only enough secondary air for complete combustion. At light loads, with certain types of coal, most burners have a tendency for coke to form and adhere to the burner tip. For this reason they should be watched closely and cleaned when needed.

As for coal fineness, it can readily be appreciated that the finer the coal, the more particles per pound, and the more particles the less the distance between each individual one together with the fact that a smaller particles takes less heat to ignite it all add up to a more stable flame at light loads.

*Detroit, Mich.* E. A. KAZMIER

## Worker's Responsibility in Industrial Safety

UP TO about 25 years ago accidents were looked upon as a natural occurrence and very little thought was given to preventing them. Many serious accidents resulted from the same hazardous conditon without it being corrected or precaution taken against future mishaps. Today most plants and shops are built with safety of man and machinery as a feature, and the workmen are reasonably educated in safety.

Even though we have developed a good record in industrial safety, we still have grounds for improvement. All accidents happen for some reason and are not just fate. If an accident is properly investigated, the cause can usually be found. Often someone has violated some natural or physical law, either through ignorance or carelessness.

For this reason I believe the key to future development of safety is principally in the hands of the workmen. We must be safety conscious and remember that what we are about to do may result in a serious accident or death to one or more fellow workers.

*Shreveport, La.* R. S. MONCRIEF.

## Education Plus

READING "Engineer Vs. Operator" in April *Power* calls to mind an incident that occurred several years ago in another state. A vacancy existed as assistant to the chief engineer, and a young man, who was a more-or-less recent graduate of a well-known technical school, was taken out of the drafting room and given charge of the boiler plant.

There were three boiler-feed pumps, only one of which was in use at one time. At more or less frequent intervals, depending on how close the water tender watched feedwater temperature, pumps would get steambound, but never caused any particular trouble. On this particular occasion the new assistant engineer happened to be on the water-tender's desk when the pump balked. He at once stepped in and undertook to remedy the trouble personally. The water tender took a comfortable seat and let him do his worst. In about 5 min. he had all three pumps steambound and every boiler in the place whistling for water. He lost his head completely and had every available man drawing fires. The upshot of the situation was a plant tied up for an hour or more.

While I heartily agree that technical education is very much to be desired, experience is a darned good teacher. A combination of the two, intelligently used, is unbeatable.

*Mechanicsville, N. Y.* R. L. LEACH

## No Room for High Hats

IT IS a very real pleasure to answer Mr. Dewees (April *Power*). First, keep mum about that $100-a-week salary of yours or you will have thousands of engineers from here landing on you who will have that berth of yours even if they have to immerse you in your own fuel-oil tanks to get it. Salaries like this for operating engineers just don't happen anymore. To hear Managements talk, $30 is far too much.

Mr. Dewees' letter is so practical, so full of common sense that it is like a bonus at Christmas time. Of course, a plant cannot be run over the telephone; by all means look around and go where the boys are supposed to go; let your men know that you are a pukka, 100% practical man yourself who has gone through the mill and cannot be fooled by careless, inefficient operation. This is all good sense but, in many cases, matters just don't go that way anymore. The practice in many large installations is to place a man on top who, given the gall and the capital to fit out an office and hire a secretary, would be a full-fledged consulting engineer. *Faut de mieux,* he is managing a power plant and the fact he is endeavoring to do it in a conscientious manner is small consolation to the operators. He is compelled to rely for advice on practical operating problems, on the person who is smart enough or lucky enough to get his ear. This works to the detriment of both management and operators unless the giver is honest and capable.

Plants intelligently operated do not have carryovers, failure of tubes or like ills; it is only in those where supervision is lax and operators inefficient or lazy where these things occur. That they do occur, however, with the frequency they do, is an indictment of present methods of hiring and training personnel. Mr. Dewees can count himself lucky that he is remote from such headaches.

Like Mr. Dewees, I feel keenly the lack of a technical education, but I can if necessary turn my hand to several other branches of the trade, whereas a technical education, unless backed by years of practical experience, is a poor overcoat when the winds of adversity blow.

Any lack of get-together spirit must be placed squarely on the shoulders of the college importations. They must learn that it is no loss of dignity to sweep a floor, mop up an oil spill or accept hints from illiterate men who are, despite lack of book learning, darned good men to have around. Fraternal acquaintance with calculus is poor help when you have lost your boiler water.

*Rutherford, N. J.* MARK BELL

## More for Silas

HERE are some answers to questions in April *Power* by Silas Dewees under the article, "Engineer Vs. Operator": "Dead-heading a boiler" as referred to in my article, "Cutting in a Boiler," in February *Power,* means carrying a boiler on the line with the fire either banked or extinguished, pressure being maintained on this boiler by leaving it cut into the header or line with the operating boilers. This is, of course, uneconomical, but is sometimes done. Except in rare cases it is unjustifiable, usually occurring in smaller plants or where supervision is slack.

There is, of course, no excuse for a person being too lazy to close a valve, and it is the fault of the supervision when such conditions are allowed, or when a person whose lack of energy interferes with the proper discharge of his duties is allowed to remain on the job. But, unfortunately, such conditions continue to exist in some plants, and I believe the majority of these cases can be traced directly to lack of supervision. There are no doubt cases where the foreman or supervisor is overworked and does not have time to be in close contact with his men. In a case of this kind he, of course, needs help or relief from part of his duties. However, there are many things which tend to increase one's lack of energy, such as unfair wages, long hours, lack of square dealing, neglected maintenance and various other things.

Having been in industrial power work for about twelve years in various sections of the country in contact with both technical and practical engineers and operators, I have found many technical men inclined to have a superiority complex when dealing with practical men. On the other hand, I have observed many "old timers" nursing a secret grudge, and sometimes a feeling of jealousy, toward technical men in general. However, I do not believe that either of these types of men could be classed as truly representative. The average man, practical or technical, must realize that both types are necessary for successful operation of industry, and that cooperation is essential.

I have found that the average technical graduate on entering public employment, particularly in the power field, realizes his limitations in not having the necessary practical experience, so is cautious in his contact with practical men. He will usually go out of his way to gain their confidence and cooperation, without which he cannot hope to succeed.

*Waynesboro, Va.* J. M. MYERS

## Why Compressed-Air Explosions

IN MARCH *Power,* C. O. Sandstrom treats us to a thesis on air-compressor fires and explosions. He cites instances of explosions that occurred shortly after starting up a compressor and suggests that idleness may have been a contributing cause. Isn't it just possible that someone started the compressor and forgot to open the cooling-water valves? I recall a compressor that was started without the cooling water being turned on and the piston rod packing on the second-stage cylinder caught fire but because it was the second stage the fire was blown outward.

In July, 1935, *Power,* a contributor said that there can be no explosion unless there is fuel. He might have added that there can be no explosion unless the fuel is in the form of either oil vapor or carbon-monoxide, for a pool of oil will not cause an explosion.

Mr. Sandstrom repudiates the idea that leaky valves will produce a temperature high enough to cause an explosion. In addition to high temperature the proper kind of fuel is necessary. I have located a leaky valve by the excessively high temperature of the valve cover, attention being drawn to it by the oil on the valve cover smoking. When we see one valve cover in a group smoking and find a badly leaking valve under it, the evidence is so convincing that I feel justified in saying that a leaky valve causes high temperature. The compressor operated at 15 lb. pressure and was fitted with spring-loaded poppet discharge valves and oscillating, mechanically operated inlet valves. A piece broken out of the valve accounted for the leak and the edges of the break were rough, with a resulting high coefficient of friction.

It is conceivable that if the temperature of a leaky valve can become sufficiently high to cause an oil film to smoke on its cover, oily matter on the inside might ignite. This could develop into a fire that would be carried into the receiver to ignite the oily coating inside. If over-pressures caused the unloader to operate and shut off the air supply while the fire is burning, it is possible that a volume of carbon monoxide might be formed. This, when air is supplied again, is all that is needed for an explosion. Such a condition can be produced in a boiler furnace with coal. Why cannot a fortuitous cycle of events produce the same condition in a compressed-air system?

The tank heads in some of the photographs shown in the article give evidence of a fire burning on their surface. If a fire was burning in the receiver, is it not possible that the temperature of the header was increased until it was too weak to stand normal pressure, and failed without an explosion occurring? The fire could start at the compressor where the highest temperatures exist and be carried into the receiver.

*Toronto, Ont.* R. MCLAREN

# READERS' PROBLEMS

## QUESTIONS for Our Readers

### Static Ruins Oilcloth

**Question 1**

WE OPERATE *a table oilcloth plant and have experienced considerable trouble with static electricity collecting small pieces of wood and grease on the surface of the oilcloth during the drying operation. We have tried several methods of removing this static, but have not yet eliminated the trouble, and will appreciate any suggestions from readers of POWER.* —M.C.D.

### Will Superheat Hurt Engines?

**Question 2**

WE HAVE *three 500-hp., 4-valve engines operating on saturated steam at 175 lb. pressure. It has been suggested that we could improve our efficiency by installing superheaters and operating the engines on superheated steam. There is sufficient room in the boilers to install enough surface for 180 deg. of superheat. I am concerned, however, about the effect of the higher temperature on the engines and shall appreciate comments concerning the operating troubles we are likely to encounter? Do any changes have to be made to the engines?*—G.W.D.

Suitable answers from readers will be paid for if space is available for publication.

## BROKEN BOILER BRACE

ANSWERS to April Question 1

**The Question**

IN OUR *boiler plant, we have several horizontal-tubular boilers. They have 18 diagonal braces from the head to the shell in each head. Recently we found one of these braces broken in the middle. No sign of distress has developed, and I would like to know if this brance can be welded. Is it dangerous to run with one brace out of 18 broken?*—H.H.R.

### Repair Immediately But Do Not Weld

THE question is too indefinite to give a true picture of the situation, for much depends on size, strength and distribution of load on the braces. Even though no sign of distress has been noted, it is dangerous to operate with one of the diagonal braces broken or cracked. The boiler is operated generally for long periods of time, therefore one cannot see whether or not any of the other braces have cracked in the meantime. The fact that one brace is already broken off shows that an added stress is thrown upon the adjoining braces. If the brace is adjacent to the knuckle of the flange it would not readily be detected from outward signs of distress, yet it would not be a safe practice to leave it.

As for welding the broken brace, *absolutely NO*. In welding, the human element enters in and the Codes are very reluctant in accepting welding unless they are sure of the ability of the welder. I note one of our largest boiler manufacturers, although welding all seams, does not weld the braces but rivets them to the heads and barrel sheets. In repair work, fusion welding (and I assume that is what H.H.R. means) shall be limited in its application to those cases where the stress is carried by other construction which conforms to Boiler Code rules and not where the safety of the boiler or vessel is dependent upon the strength of the weld.

You will note that in Boiler Codes where braces are welded that the welded portions of braces have their strength cut considerably. I believe no boiler inspector would allow the boiler to operate, although much depends upon conditions covering the distribution of the stays and the load on them.

*Binghamton, N. Y.* C. W. CARTER, JR.

### Foolish to Operate With Broken Stay

DIAGONAL stays are placed in a boiler to impart strength and not ornamentation, therefore it is foolish to operate a boiler which has a broken stay. Brace spacing and size are calculated to withstand the load so that each brace is under equal load. When one stay breaks, undue stress is placed upon the rest and, although no trouble has as yet developed, that does not signify that grave trouble will not be experienced later. If the broken stay is in good condition it may be welded by a skilled welder. If it shows signs of deterioration due to pitting or corrosion, it would be advisable to install a new one. If the stay is welded it should receive particular attention during each internal inspection of the boiler.

*New York, N. Y.* J. W. EDWARDS

### Consider Future Danger

THE broken brace is a sign that distress has developed, probably caused by unequal expansion of the tubes. The extra strain will not be divided proportionately over the other 18 braces, as each brace was pitched and designed with a factor of safety sufficient to resist pressure over its own particular area. According to the number of braces, it can be presumed that they are of the rod or palm and not angle, "T" or gusset stay type.

The boiler is not in a dangerous condition, but in future, and until this break has been taken care of, there will be that uncomfortable feeling among the personnel concerned. If other boilers are available, do not continue to operate this one. The next break will further lower the factor of safety and do at least twice the damage.

It must be taken care of. The opinion of a reliable welder should be accepted in regard to the efficiency of a welded section, probably to be done under unfavorable conditions. Instead, the removal of four rivets and a new brace, riveted, will prove worthy of the extra trouble and expense.

*New York, N. Y.* D. ELLIS

### Calculate New Safe Working Pressure

H.H.R. should not attempt to weld the broken brace. Welding will not restore the brace to the original length or tension but will tend to increase the length and thereby cause a slack brace.

A boiler maker should be called in at the same time that the boiler inspector is there and a close inspection made of the other braces. The boiler maker would know the proper length to make the new brace to secure proper tension when it is riveted in place. The usual method is to have the brace slightly shorter than the distance from the head to the center of rivet holes in the shell. The brace is then riveted to the head and heated until it lengthens enough for the rivet holes in the brace to line up with rivet holes in the shell.

The boiler should not be operated with a broken brace unless the pressure is enough below the S.W.P. to insure no overloading of the remaining braces.

The simplest way to determine the safe pressure to carry with the brace broken would be to multiply the S.W.P. by 17/18.

*Dorchester, Mass.* W. F. O'REGAN

### Repair, But Do Not Weld

THE brace could be welded but it would be inadvisable to do so. In fact, in districts where a boiler code is in force, it would not be permitted. In this case the logical thing to do would be to put in a new brace. In the operation of any boiler, safety should be the first consideration, and I consider it dangerous to carry on with a broken stay.

*Montreal, Que.* NICHOLAS FORMBY

### Rod-Type Braces Can Be Welded

WHILE it is possible that a boiler with 18 diagonal braces from the head to the shell in each end could be safely operated with one broken, the strain would not be equally distributed, and it would be better to make the necessary repairs by either replacing or welding the brace. If the braces are pressed steel, it would be better to take out the broken brace and have a new one made. If they are of rod type, it can be welded and a good job obtained, if provision is made for expansion and subsequent shrinkage.

To take care of this expansion and shrinkage, remove the bolt from the bifurcated end of the brace, where it is attached to the tee-iron on the boiler head, and substitute a wooden dowl or a soft-metal plug. Line a piece of light angle iron with asbestos paper

and wire the brace into it lightly, making the wire tight enough to hold the brace in line until the weld is well under way, after which the angle iron may be removed for finishing. Allow the brace to cool thoroughly, then remove the wood dowel or soft plug, and ream the bifurcated end of the brace and the tee-iron to receive a slightly oversize bolt. This reaming will line the holes to receive the bolt with a solid setting.

The purpose of using a wood dowel or soft-metal plug is to allow expansion and shrinkage to be taken up at that point, rather than throwing undue stress into the brace itself. The reaming is to neutralize any differential which may have occurred in the welding, resulting in misalignment of the holes in the bifurcated end of the brace and in the tee-iron. Unless such provision is made, a brace of this nature cannot be successfully welded, because it will be under intense strain after cooling, and will soon break again.

*Peoria, Ill.* JOHN E. HYLER

## CHANGING DIESEL PISTON RINGS

ANSWERS to April Question 2

### The Question

WE HAVE *a 300-hp. diesel that has been giving us only 1,000 rated hp.-hr. per gal. of lubricating oil. Our belief is that rearrangement of the piston rings will reduce oil consumption without detriment to engine operation. These pistons now have four rings at the crown. Our belief is that a wiper ring on the skirt and a wiper ring replacing the lower ring at the crown will solve the problem. Is this logical, and if so, will it require any change in lubrication methods?*—A.V.M.

### Change Rings In Vertical Engine

A.V.M.'s question leaves many factors unstated. However, if excessive oil consumption occurs in a vertical engine, and there are no other contributing causes, such as pistons or cylinders being worn out of round, and there are (as factory equipped) no oil or wiper rings, a change in rings alone will work wonders.

It is not necessary to install an extra wiper ring in the piston skirt if the lower ring groove in the crown is properly drained or vented to allow excess oil to flow into the piston. A wiper ring at this point will remove much of the excess oil; though if the oil consumption still shows higher than normal, removal of the lowest of the three remaining compression rings and its replacement with a wiper ring, after drilling oil drains through the piston wall to carry away excess lubricant, will eliminate the surplus.

There are many old diesels driving pipeline pumps in the East Texas field which have had their lubricant consumption halved by following the plan outlined above. The sole upkeep or maintenance needed is to pull the pistons every three months to clear the oil-groove drain holes of accumulated carbon.

If the diesels are using suitable oil it is most likely the rings are to blame; but if, as frequently happens, some *price*-favored oil is being used, without too close an inquiry into its suitability, a change to some recognized brand might clear up the trouble without any mechanical change in the engines. In one plant which came under my observation, the change from an oil of approximately S.A.E. 40 specifications to one more nearly S.A.E. 30 reduced oil consumption in six 400-hp. diesels by 23% without any other change.

*Longview, Texas* ELTON STERRETT

### Make Oil Scraper From Bottom Ring

TO PREVENT oil pumping, round off the corners of the first three rings on the upper corners. The bottom ring can be made into an oil scraper or one may be purchased. To make the old ring into an oil scraper will require beveling the face of the ring from the top about halfway down. The lower edge of the ring groove should be beveled not to exceed $\frac{1}{4}$ of its depth. From this bevel, $\frac{3}{32}$-in. holes are drilled to the inside of the piston at a 45-deg. downward angle. Space holes at equal distances around piston.

On some pistons, rings can not be installed at the bottom because the piston skirt overtravels the lower cylinder bore. If these pistons do not, then a ring groove can be turned and a special oil ring installed.

After these changes A.V.M. should cut down on the cylinder feed and watch his pistons until he gets the correct amount without forming a lot of sticky black carbon. More pistons are hurt by the use of too much oil than by running without the proper amount. Few engineers are so saving that they starve the pistons.

Too much oil causes carbon which builds up the piston friction, causing the operator to think that he is not feeding enough oil. So round and round it goes until we have very poor lubricating oil economy and a dirty job of cleaning carbon and loosening rings.

*Walthill, Nebr.* WM. W. DINGWALL

## CORRECTIONS

On the portion of the psychrometric chart used to illustrate mixture calculations, page 189, of the April section on Air Conditioning, a part of the scale of total heat is incorrectly labeled and indicates a heat content 1 B.t.u. too high between 10 and 26 B.t.u.

In the article describing the new Lafayette, La., diesel plant, page 201, April *Power*, the ninth line of the first paragraph incorrectly quotes fuel cost as 3.6 cents per kw.-hr. This should of course be 3.6 mills as indicated by the context.

In the description of the Millers Ford Station boiler, May *Power*, page 262, the ash-fusing temperature of the fuel to be burned is given as 2,000 deg. F. It should instead be 2,600 deg. F.

### More Explanation On Feedwater Treatment

I DO not think that Wm. J. Ryan adequately answers the question of A.E.T. in March *Power*. The important question for A.E.T. is not whether he should start with $\frac{1}{2}$ lb. or 3 lb. per 1,000 gal. but whether the water is properly treated. The alkalinity test will not tell him this.

Even when the constituents of the raw water are exactly known, calculation of required chemicals is good simply as a starting point. Subsequent feed of chemicals must be regulated by tests of the boiler water.

My guess is that the chemist who advised 3 lb. is wrong and that the $\frac{1}{2}$ lb. recommended by the supplier of the chemical is right. I believe the chemist based his calculations on the chemical action of sodium carbonate and trisodium phosphate, but ignored the mechanical action of tannin. A small amount of tannin greatly reduces the amount of chemicals required.

It is not very important whether A.E.T. starts with $\frac{1}{2}$ lb. or 3 lb., but my advice is that he start with $\frac{1}{2}$ lb., then test the water for alkalinity, causticity and hardness, and adjust the amount of chemicals according to the hardness and causticity. The alkalinity is of no value as a guide for internal treatment with compound.

I would advise A.E.T. to proceed thus: Start with $\frac{1}{2}$ lb. per 1,000 gal. and test water for hardness and causticity. If hardness is more than 2 grains (per gallon) and causticity less than 5 grains, increase the compound; as long as hardness is more than 2 grains and causticity less than 5 grains there is a deficiency of compound. If the causticity is over 15 grains, though the hardness is not below 2 grains, the compound should not be increased.

When water is internally treated with compound, correct treatment is obtained when hardness is between 1 and 2 grains and causticity between 5 and 15 grains. If the causticity is over 15 grains the compound should be reduced until the hardness begins to rise. If the hardness can be maintained around 1 grain with less than 5 grains of causticity, that is quite all right; in the usual case higher causticity will be needed to obtain a low hardness.

Since A.E.T. may not know how to test for hardness and causticity, the method is: *Hardness.* Place 58.3 cc. of boiler water in a 250-cc. bottle. Add soap solution and shake until the lather formed will remain unbroken for 5 min. With a standard soap solution the cc. of solution used equals the hardness in grains of calcium carbonate per gallon.

*Causticity or Ph alkilinity.* To a 58.3-cc. sample of boiler water add 2 or 3 drops of phenolphthalein indicator to turn the water pink, then (while stirring) add N/50 standard sulphuric acid solution until the pink

disappears. The cc. of acid used equals the causticity as gr. per gal. of calcium carbonate. To get total alkalinity, add to the same sample 2 or 3 drops of methyl orange indicator to color the water yellow. Then (while stirring) add the N/50 sulphuric acid and stir until the color changes to orange. The amount of sulphuric acid used in the previous test plus the amount used in this test is the total alkalinity as gr. per gal. of calcium carbonate.

*Chicago, Ill.* N. T. PEF

[*The foregoing was submitted to Mr. Ryan for comment. His remarks follow. —Editor*]

It is true that the presence of tannin will reduce the amount of reacting chemicals below the amount otherwise required.

The hardness test described by Mr. Pef is correct.

The test he gives for methyl orange alkalinity is the same as that given in March *Power*, except that the previously described test gives the alkalinity in parts per million, whereas Mr. Pef's test gives grain per gallon (17.1 p.p.m. equals 1 gr. per gal.).

The test for causticity described by Mr. Pef does not give the causticity, but gives the phenolphthalein alkalinity, the value of which is affected by both the monocarbonate alkalinity and the hydroxide alkalinity (causticity).

If reliance is placed on the mechanical action of organic materials in a boiler compound, the supplier of the material should prescribe what tests to make, as he knows what is in the compound and what it is supposed to do.

When using a straight inorganic material, such as soda ash (sodium carbonate), a certain excess of the chemical is maintained in the boiler water. The amount of carbonate present can be computed from the alkalinity tests as follows:

If phenolphthalein alkalinity is one-half methyl-orange alkalinity, or less, the carbonate present (as $CaCO_3$) is equal to twice the phenolphthalein alkalinity. No causticity will be present.

If the phenolphthalein alkalinity is greater than one-half the methyl-orange alkalinity, which is usually the case in boiler water, the carbonate will be equal to twice the difference between the total alkalinity and the phenolphthalein alkalinity. The causticity will then be equal to twice the phenolphthalein alkalinity minus the methyl-orange alkalinity.

Maintaining a carbonate alkalinity of 30 p.p.m. or higher should prevent hard scale deposit.

If a certain concentration of phosphate is to be maintained in the boiler water, a simple test method is to use color comparison standards. Certain reagents are added to a sample of the water, and the development of color is in proportion to the amount of phosphate present.

*New York, N. Y.* W. J. RYAN,
*Water Service Laboratories*

# MANUFACTURER'S BULLETINS

**Motors**—Allis-Chalmers Mfg. Co., Milwaukee, Wis. 4-page illustrated leaflet No. 2173-A on Type AR squirrel-cage induction motors. Cutaway illustrations.

**Protection**—Chicago Eye Shield Co., 2300 Warren Blvd., Chicago, Ill. 20-page catalog of industrial head and eye protection devices.

**Disconnect Switches** — Delta-Star Electric Co., Chicago, Ill. 2-color, 24-page bulletin describing "PM-22" and "PM-23" group-operated pole-top disconnecting switches in ranges from 7.5 to 161 kv. Complete engineering data.

**Water Analysis**—W. H. & L. D. Betz, Philadelphia, Pa., 235 West Wyoming Ave. 30-page second edition of "Water Analysis, Methods, Chemicals, Apparatus."

**Spray Nozzle**—Chain Belt Co., 1600 West Bruce St., Milwaukee, Wis. 4-page Bulletin No. 278 on "Rex" non-clogging, flat spray nozzles.

**Steam Turbine Controls**—Moore Steam Turbine Corp., Wellsville, N. Y. Two sheets on steam-turbine controls for auxiliary drives.

**Superheater**—Controlled Heat Equipment Co., Dime Bank Bldg., Detroit, Mich. 16-page booklet describes and illustrates water-regulated superheater.

**Turbine-Generator Sets**—General Electric Co., Schenectady, N. Y. 4-page Bulletin GEA-2295, on d.c. sets of from 10 to 400 kw.

**Switchgear**—General Electric Co., Schenectady, N. Y. Single sheet GEA-2309 on switchgear panels for engine-driven generators.

**Thermometers**—Bristol Co., Waterbury, Conn. 88-page catalog No. 1250 covers liquid-filled, vapor-tension, gas-filled recording, indicating and controlling thermometers.

**Air Compressors**—Sullivan Machinery Co., Dept. 21, Michigan City, Ind. Bulletin 88-0 on two-stage, air-cooled, V-type, single-acting, stationary, air compressors.

**Crushers** — Jeffrey Mfg. Co., Columbus, Ohio. 20-page illustrated catalog No. 621 on single-roll crushers and 8-page illustrated bulletin No. 622 on double-roll crushers.

**Temperature Control** — The Bristol Co., Waterbury, Conn. 6-page leaflet No. 443 on automatic temperature control.

**Hoist**—Harnischfeger Corp., Milwaukee, Wis. 4-page, 2-color illustrated Bulletin H-1 describes "Zip-Lift" 250 and 500-lb. hoists.

**Non-Return Valves** — Edward Valve & Mfg. Co., Inc., East Chicago, Ind. 26-page illustrated catalog No. 11-B covers non-return valves, stop check valves in sizes from ¼ to 2 in., with screwed and flanged ends. Discusses use of welding ends and motor controls.

**Air Compressor** — Allis-Chalmers Mfg. Co., Milwaukee, Wis. 4-page illustrated leaflet 2218 describes "Ro-Twin" for up to 100 lb.

**Metallic Cement**—Smooth-On Mfg. Co., 568-574 Communipaw Ave., Jersey City, N. J. 32-page pocket-size manual.

**Pump for Air Conditioning** — Ingersoll-Rand Co., 11 Broadway, New York, N. Y. 6-page profusely illustrated bulletin No. 2200 describes Cameron motorpump for air conditioning.

**Diesel Engines**—Hercules Motor Corp., Canton, Ohio. 4-page Bulletin E-136 gives specifications, installation diagrams and power charts for Models DJXB 3¼x4½-in. and DJXC 3¼x4½-in., 6-cyl. diesels.

**Metal Replacement** — Gamlen Chemical Co., 61 Fremont St., San Francisco, Calif. 4-page leaflet on "Gamlenite Metal Replacement" for boiler heating surfaces.

**Speed Control** — Reeves Pulley Co., Columbus, Ind. 12-page illustrated bulletin No. T-361 on "Automatic Production Control."

**Voltage Control** — Allis-Chalmers Mfg. Co., Milwaukee, Wis. 12-page illustrated bulletin 1176 on electrically-energized grids for controlling mercury-arc rectifiers and inverters.

**Water System** — Roots-Connersville Blower Corp., Connersville, Ind. Single sheet Bulletin 260-B13A covers 300 and 500-gal. per hr. systems for shallow wells and cisterns.

**Portable Scraper Conveyor**—Jeffrey Mfg. Co., Columbus, Ohio. 4-page illustrated bulletin No. 623 describes Type 219-B portable scraper conveyor.

**Water Softening** — Permutit Co., 330 West 42d St., New York, N. Y. 48-page booklet No. 592 on hot lime soda water softening equipment for boiler feed-water conditioning.

**Water Heaters**—American District Steam Co., North Tonawanda, N. Y. Bulletin No. 3576 on copper U-tube water heaters.

**Portable Cable** — General Electric Co., Schenectady, N. Y. 28-page bulletin GEA-1918A on tellurium, all-rubber, glyptal compound and braided types of portable cable.

**Condenser Tubes** — Scovill Mfg. Co., Waterbury, Conn. Price list on condenser and heat exchanger tubes.

**Circuit Breaker**—Condit Electrical Mfg. Co., Hyde Park Station, Boston, Mass. 4-page Section No. 521 describes Type DZ-40, 100,000-kva. station-type air circuit breakers.

**Steam Engines**—Troy Engine & Machine Co., Troy, Pa. 48-page illustrated bulletin No. 305 describes and shows applications for "Modern Steam Engines." Discusses by-product power, engine-generating sets and horizontal and vertical steam engines.

**Combustion Meters**—Hays Corp., Michigan City, Ind. 14-page catalog RA-346 on indicating and recording combustion meters.

**Welding Brass and Bronze**—Linde Air Products Co., 30 East 42d St., New York, N. Y. 12-page, 8½x11-in. illustrated booklet No. F.2832A on technique of fusion welding brass and bronze and welding of commercial yellow brass pipe.

**Mortars and Plastics**—Babcock & Wilcox Co., 85 Liberty St., New York, N. Y. 8-page illustrated bulletin on refractory mortars and plastics.

**Boiler**—Babcock & Wilcox Co., 85 Liberty St., New York, N. Y. 24-page, 2-color, illustrated Bulletin G-17-A, describes integral furnace boiler.

**Aftercooler** — Ingersoll-Rand Co., 11 Broadway, New York, N. Y. 4-page Bulletin No. 2239 on "Pipe-Line" Aftercoolers.

**Leather Belting**—E. F. Houghton & Co., 240 West Somerset St., Philadelphia, Pa. 40-page manual "How to Belt Your Drives for More Profit."

**Heat Exchangers**—Sims Co., P. O. Box 1096, Erie, Pa. 8-page well illustrated bulletin describes diesel exhaust gas and jacket-water heat exchangers.

**Insulating Firebrick**—Babcock & Wilcox Co., 85 Liberty St., New York, N. Y. 8-page illustrated bulletin gives complete information on insulating firebrick.

**Corrosion Resistant Compounds** — Alox Corp., P. O. Box 949, Niagara Falls, N. Y. 2-page mimeographed Bulletin C-36 on use of Type "FFK" corrosion-resistant compound.

**Mechanical Transmission**—Ohio Gear Co., 1333 East 179th St., Cleveland, Ohio. 128-page catalog describes line of spur, bevel, worm and other gears and motorized speed reducers.

**Refractories**—Sanford C. Smith Refractories, Inc., 1715 Niagara St., Buffalo, N. Y. 12-page booklet describes eight types of "Flame Brand" high-temperature refractories.

**Diesel Power**—Thermal Engineering Co., Boston, Mass. 12-page bulletin on "Tecomatic" diesel power describes system of automatic operation of small diesel plants.

**Tank Heater**—Titusville Iron Works Co., Titusville, Pa. 4-page bulletin on suction tank heater.

*(Additional Bulletins on Page 56)*

# THE ENGINEER'S BOOKSHELF

## Rectifier Practice

MERCURY ARC RECTIFIER PRACTICE (1936)—*By Frederick C. Orchard, A.M.I.E.E. Printed and published in England, distributed in America by Instruments Publishing Co., 1117 Wolfendale St., Pittsburgh, Pa. 224 pages, 6x9 in. Cloth. 106 illustrations, many tables. Price $5.*

While British and American rectifier theory are the same, the equipment is somewhat different. This book gives a complete picture of British practice. There are eight chapters, General Principles, 33 pages; Construction of Rectifier Plant, 30 pages; Rectifier Substations, 32 pages; Rectifier Plant, Installation, 19 pages; Sub-station Maintenance and Operation, 20 pages; Grid Control, 26 pages; Conclusion, 12 pages. An appendix contains tables of Maximum Safe Loads on Chains and Wire and Hemp Ropes, Standard Copper Conductors and Fuse Wire.

## Economics

PLATFORM FOR AMERICA (1936)—*By Ralph E. Flanders, past president, A.S.M.E.; president Jones & Lamson Machine Co. Published by Whittlesey House, McGraw-Hill Book Co., Inc., 330 West 42nd St., New York, N. Y. 118 pages, 5½x8¼ in. Cloth. Price $1.*

Problems facing business today—government on one side, consumer on the other—are here analyzed. Mr. Flanders argues that we should not produce less, but more; and that prosperity and plenty for all will be brought about by governmental encouragement of large profits rather by limitation of them. He shows the community of interest between agriculture, business and labor, and shows that no policy or legislation which benefits one group at the expense of another will, in the end, benefit the country as a whole. It is stimulating reading, if you are national-minded.

## Diesels

DIESEL HANDBOOK (3RD EDITION)—*By Julius Rosbloom. Published by Diesel Engrg. Institute, 443-445 Hoboken Ave., Jersey City, N. J. 544 pages, 5 x 7 in. Fabricoid covers. Price $5.*

The annual revision of this extensively illustrated handbook, including a little of everything from micrometer practice to fundamental heat laws, plus a lot of tips and hints for the diesel operator. Language is occasionally a little confused, but probably the variety of material balances that.

## BRIEF REVIEWS

EYESIGHT PROTECTION. (1936) *Six bulletins published by National Society for Prevention of Blindness, Inc., 50 West 50th St., New York, N. Y. Publication No. 188,* "EYE EXAMINATION IN INDUSTRY." *12 pages, 5 cents. Publication No. 77,* "EYE PROTECTION IN INDUSTRY," *8 pages, price 5 cents.* "GOGGLES DO SAVE SIGHT," *single sheet. Publication 173.* "THE CO-OPERATION OF THE PHYSICIAN AND THE SAFETY ENGINEER IN SAVING SIGHT." *12 pages, price 5 cents. Publication 87,* "GOOD EYESIGHT IN INDUSTRY," *4 pages, price 5 cents. Publication 48,* "EYE ACCIDENTS IN INDUSTRY," *48 pages, 6x9 in., price 25 cents.*

CONDENSED LIST OF SOURCES OF INFORMATION ON AIR CONDITIONING (1936). *Published by the Electrical Division of the Bureau of Foreign and Domestic Commerce, Dept. of Commerce, Washington, D. C. 16 mimeographed pages, 8 x 10½ in. Paper. Free.*—Condensed from Third Edition of "Bibliography of information on Air Conditioning.

RESEARCHES IN HIGH-PRESSURE STEAM AT PURDUE UNIVERSITY, 1931-1935 (1936), *by A. A. Potter, H. L. Solberg, and G. A. Hawkins. Published as Research Series No. 52 by Engineering Experiment Station, Lafayette, Ind. 22 pages, 6 x 9 in., paper. One diagram and one chart. Limited number free.*

RESEARCH FOR INDUSTRY (1936). *Published by Ohio State University, Columbus, Ohio. 8 pages; 8½ x 11 in.; many illustrations. Paper. Free.*—Facilities for industrial research offered at Ohio State University.

SUMMER SESSION AT PURDUE UNIVERSITY (1936). *Vol. 36, No. 6 of Bulletin of Purdue University, West Lafayette, Ind. 64 pages, 6 x 9 in.; paper. Free.*—Catalog of summer session at Purdue from June 11 to Aug. 8, 1936.

# EXHAUST LINES

### *By E. J. Tangerman*

Tomato plants so large the fruit must be picked with a stepladder and yielding 217 tons per acre instead of the average 5 tons, potatoes at 2,465 bushels per acre (without real pushing) instead of the average 116, and tobacco plants 22 ft. high with all leaves "center leaves"! Sounds like Baron Munchausen or some South Sea island, but it has been done in these United States by Dr. W. F. Gericke, University of Calif. But the acres that do this phenomenal producing are not out on the ground—they're in tanks, with the plant seeds held in a thin bed of excelsior, peat, or moss on a wire screen above a bath of heated nutrient liquid. Roots dip into the liquid, which is kept at proper temperature by soil-heating cable. Why mention it here? Because where there's heat there must be power or steam, and that's where we come in.

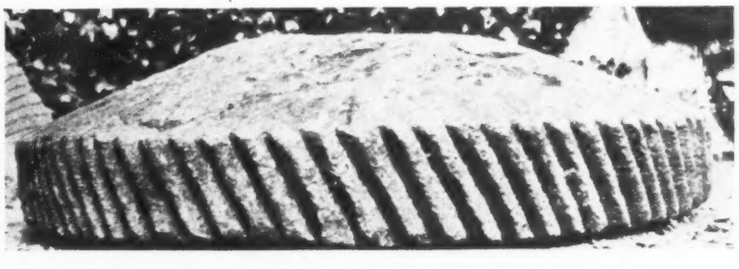
*J. L. Chaloner, well-known British diesel engineer, predicted recently before the Diesel Engine User's Assn. of London that within four years, England will have replaced 200,000 gasoline vehicles by the same number of "oilers," or diesel and semi-diesel units.*

One of the recent applications of metal spray was to "salt" a gold mine. If you haven't heard the term, or haven't invested in a mine like that, it's an unscrupulous man's trick for planting bits of gold in ore or sand, then showing the treated spots to a sucker and getting him to think the whole area is just like the sample. In this case the salter was particularly careful—he sprayed his gold through a 200-mesh screen to be sure it was fine enough. Which proves one thing, anyway—that the sprayed particles are smaller than 200-mesh.

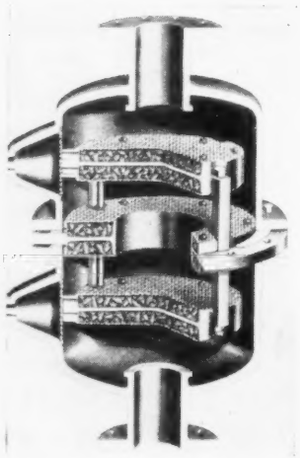
*Bituminous coal burned "raw" is extremely wasteful as well as smoky. Now three scientists, Karrick, Jacobsen and Carter, out at the University of Utah, are testing conversion of bituminous into solid smokeless fuels, artificial 1,000-B.t.u. gas, or crude oil for high-octane gasoline. Cost of treating runs only $1 to $1.25 a ton.*

A subterranean periscope has been developed at Norris Dam. With it, engineers can inspect foundation rock forty feet under ground. Brass pipe and electric light are run down a drilled hole.

*Long before Columbus discovered America (or did he?), a Swedish community had its own grain crusher driven by helical gears. As shown in the photograph, they looked a lot like present-day helicals, but were cut eight or nine centuries ago. Otto Lundell, president of Michigan Tool Co., Detroit, found them in the ruins of an old Swedish castle. And whoever made them knew what he was doing—his design is in accord with modern engineering practice. Notice particularly how the gear is thicker at the hub than at the teeth to provide adequate bearing area.*



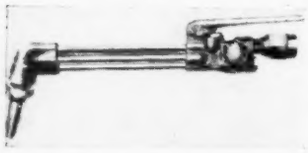
# WHAT'S NEW IN PLANT EQUIPMENT



### PURIFIER AND DRIER

"EXHAUSTEAM" purifier and drier for exhaust steam, air and gases subjects vapor to sudden reduction in velocity, sudden deflection by baffling, centrifugal force and capillary action, thus using several methods of purifying and drying.

*Morehead Mfg. Co., Detroit.*



### OXY-ACETYLENE CUTTING ATTACHMENT

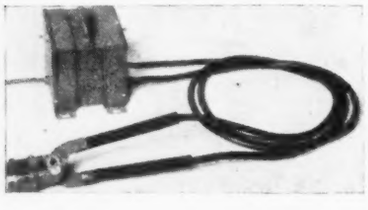
OXWELD Type CW-22 welding attachment for Types 17 or 22 blowpipe handles is suitable for all but heaviest work. Cutting valve centralized under cutting lever has removable seat. Mixed gas passage of three Ambrac tubes. Forged-bronze body with forged, manganese-bronze head.

*Linde Air Products Co., 30 East 42nd St., New York, N. Y.*

### ELECTRIC PLIERS

THERMO-GRIP pliers have renewable carbon blocks in ends of jaws which carry large currents at low voltage from transformer to sides of fitting to be heated. Uses include soldering, sweating joints, etc. Cast-bronze pliers have 2-position opening and fibre handles. Available for 110 volts, 25, 50-60 cycles and 220 volts 50-60 cycles.

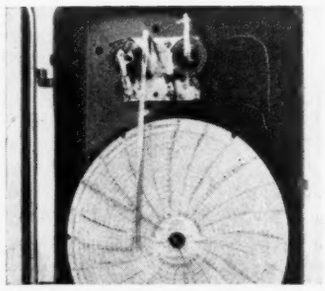
*Ideal Commutator Dresser Co., 1025 Park Ave., Sycamore, Ill.*



### GAS-FILLED RECORDING THERMOMETER

SMALL-BULB, gas-filled thermometer, for —60 deg. F. to 1,000 deg. F., where tubing as long as 100 ft. or more is required, has compensating device to correct for temperature changes in case and tubing. Rectangular case, one or two pens with 12-in. chart, one with 8-in. size. Catalog 1250.

*Bristol Co., Waterbury, Conn.*



### COMMUTATOR AND SLIP RING GRINDER SUPPORT

TYPE O.E. grinder support attached to motor so resurfacing element is on rigid and inflexible base between brush-holders; allows resurfacing without dismantling brush-holders or rigging. Grinding done at normal operating speed with motor in its own bearing. Type T.O. support for attaching to bus yoke and bearing pedestal. W type for mounting Ideal grinders on larger exciters, motors and generators. Attaches to main frame of machine. Universal type attaches to main housing and bearing support. Plate type for smaller units attached to end bell of motor. Bulletin GB-735.

*Ideal Commutator Dresser Co., Sycamore, Ill.*

### VAPOR-SPRAY CLEANING UNIT

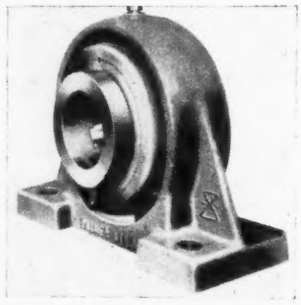
MODEL G, "Hypressure Jenny," sprays combination of water vapor, hot water and cleaning chemicals at 50 to 150 lb. per sq.in. and costs half as much as older type. Vertical-spiral-parallel-coil vapor generator controlled by automatic switch. Burns kerosene or light furnace oil. Single triplex plunger pump has mechanical inlet check valves. Stationary or portable.

*Homestead Valve Mfg. Co., Coraopolis, Pa.*

### LUBRICATING OILS

SIX grades of oil designed to replace large number of special types of lubricating oils to simplify selection and use. Said to have higher stability, good demulsibility, low carbon-forming tendency, good viscosity index and color and low pour test.

*Standard Oil Co., (Ind.) 910 S. Michigan Ave., Chicago, Ill.*



### SELF-ALIGNING PILLOW BLOCK

TO SIMPLIFY alignment of shaft and take care of light or heavy shock loads at either high or low speed. Cast-steel housing and ball and socket bearing. Housing is machined to spherical surface and fit that permits free oscillating movement of about 3 deg. in any direction.

*Palmer-Bee Co., Detroit, Mich.*

### RATIO DIFFERENTIAL RELAY

TYPE HA relay protection of a.c. apparatus has operating time of one cycle. Two types, for generator and motor protection, and one for transformers. Generator and motor relay operate on internal troubles with 5% of smaller secondary current without tripping on heavy overloads or on external faults. Provided with taps to operate on unbalance equal to 5, 10 or 20% of smaller secondary current. 5½ in. square. Relay for protection of two winding transformers has taps so that current balancing transformers are unnecessary and will operate on fixed differential of 25%. Tripping suppressor prevents operation on transformer magnetizing current inrushes. Relay is 5½ x 12 in.

*Westinghouse Electric & Mfg. Co., East Pittsburgh, Pa.*

### LIGHT-DUTY BELT

BELTPACO JUNIOR transmission belt for light or fractional-hp. drives may also be used as shaping or forming belts for light conveyors. Extra plies for "body," yet flexibility allows belt to hug pulleys.

*New York Belting & Packing Co., 1 Market St., Passaic, N. J.*

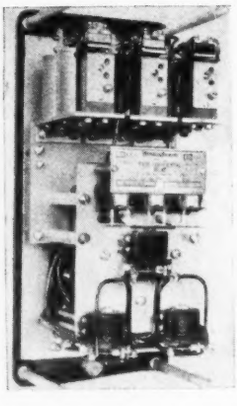
### SQUIRREL-CAGE MOTORS

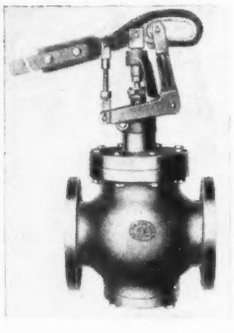
SPLASH-PROOF, squirrel-cage motors in all ratings from ½ to 200 hp. for constant-speed and multi-speed continuous or intermittent duty in all voltages and cycles for any torque and starting current. Construction provides protection for inner parts from splash or spray, yet permits proper ventilation by small openings in bottom of end brackets for air intake and in frame cover under motor for exit.

*Imperial Electric Co., Akron, Ohio.*

### ADJUSTABLE FULCRUM FOR BALANCE VALVES

SELF-ADJUSTING fulcrum provides increasing or decreasing movement of valve disk from and to seat for fixed movements of lever. Disk movement is least when near seat or on load requirements and increases as load increases. Power increased by having fulcrum at shortest point when valve disk is





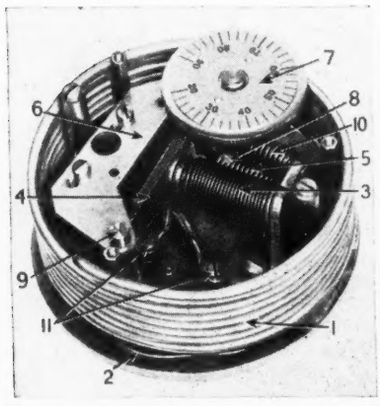
being lifted from seat. Single- or double-seated types with or without removable-cage-type interiors.

*Ruggles-Klingemann Mfg. Co., Salem, Mass.*

### INSTANTANEOUS ELECTRIC ROOM THERMOSTAT

TYPE LSB thermostat for unit heaters and coolers connects to lighting circuit, and on a.c. can handle up to 1,000 watts without relay. Long sensitive bulb (1) connecting tube (2) and flexible bellows (3) are filled with temperature-sensitive liquid. Temperature rise causes pressure which expands bellows, pushing lever (4) up, expanding loading spring (5). Lever contacts pin of Micro switch (6) clicking it on or off as required. Adjustment made by turning dial (7) partly visible and adjustable, through protective covering. Dial shaft carries cam (8) which moves lever (4) closer to or farther away from bellows (3) thus varying temperature at which contact with switch occurs. Switch mechanism on pivot (9) held by spring (10). Lead wires carried through case at (11) to terminal block on back. Will operate on bellows movement of 0.001 in.

*Wilbin Instrument Corp., 40 East 34th St., New York, N. Y.*




### PORTABLE CABLE ACCESSORIES

FOR coupling cable to portable electric equipment and to connect lengths of cable, high-voltage couplers, rated at 100 amp. are particularly adapted for G.E. 2,500-, 3,500- and 5,000-volt, Types SH and G, portable cable. Shielding tapes or ground conductors can be brought out of coupler socket for grounding. Malleable-iron socket and plug have covers to keep out dirt and moisture. Sliding-spring type contacts are enclosed and shielded by tubing.

Molded-rubber terminals for terminating portable or drag cable holds single conductors together. Made of tellurium compound, and provides for grounding of wires and braided shields.

*General Electric Co., Schenectady, N. Y.*

### BELT FOR SMALL PULLEYS

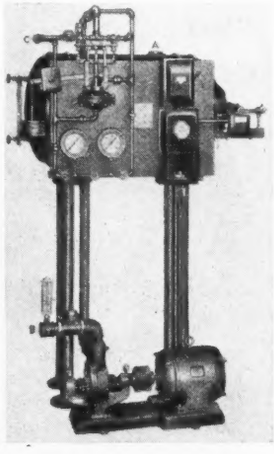
PLYTEX belting for high-speed transmission over small pulleys has light-weight silver duck fabric. Furnished in 4, 5, 6, and 7 plies from 1 to 8 in.

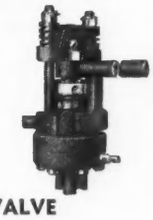
*Thermoid Rubber Co., Trenton, N. J.*

### DIFFERENTIAL DRAINAGE BOILER-RETURN SYSTEM

CLOSED-TYPE pumping system returns condensate to boilers at temperatures above 300 deg. F. Claimed to save 15% over trap systems and to cut pressure differentials, thus saving wear and tear on traps and rest of system. Consists of return tank and steam pump with necessary gages and valves for automatic operation. A—Return line from steam traps; B—Feed line to boiler; C—Initial pressure to steam units. Bulletin No. 50.

*Stickle Steam Specialties Co., Indianapolis, Ind.*





### HYDRAULIC VALVE

STYLE "H," balanced, flat-disk 2, 3 and 4-way hydraulic valves for oil or water up to 5,000 lb. pressure act like differential pistons. Pressure in body of valve presses disk down and smaller-area stem up. Two spring adjusting nuts exert required downward pressure on stem through crossbar and ball. Cast-steel or bronze body with Monel stem, stainless-steel base and steel lever. Bulletin 236.

*W. H. Nicholson & Co., 12 Oregon St., Wilkes-Barre, Pa.*

### DUPLEX RUBBER HOSE

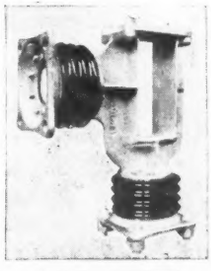
SUPERO Siameez twin rubber hose for gas welding and cutting, etc., has connecting web molded between and cord-wound reinforcement between first and second braid. ¼-in. size has bursting strength of 2,000 lb. per sq.in. Hoses may be same or different colors. Connecting web cut with Carborundum wheel for attaching hose to separate outlets.

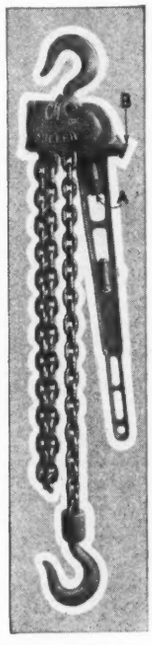
*Electric Hose & Rubber Co., Wilmington, Del.*

### BUS SUPPORT

HEAVY-DUTY, 7.5-kv. bus support to carry eight 8x¼-in. copper bars, constituting main bus of large steel mill. Lower insulator mounted on steel I-beam, side-supporting insulator on concrete wall. Supports equipped with rollers and spring spacers permitting free lateral movement.

*Delta-Star Electric Co., 2400 Block, Fulton St., Chicago, Ill.*






### PULLING TOOL

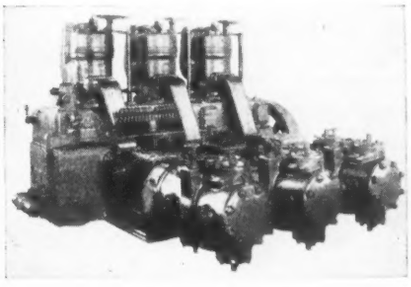
PULLS at any angle with capacities of ¾ and 1½ tons. Gear reduction allows 43 lb. on ratchet handle to pull ¾ ton. ¾-ton size weighs 17 lb. Minimum distance between hooks on ¾-ton size is 9 in. Collapsible handle for tool box storage. "Herc-Alloy" double-duty alloy-steel chain. Operation controlled by two buttons, which permit fine adjustment of chain.

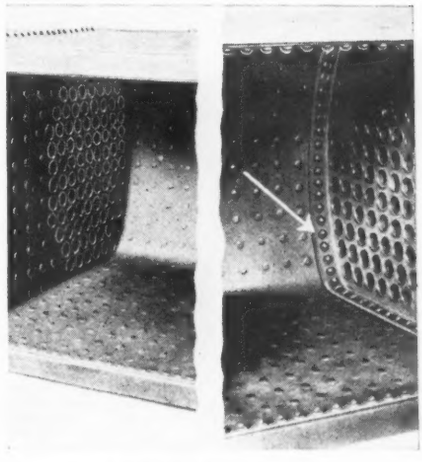
*Chisholm-Moore Corp., Tonawanda, N. Y.*

### RIGHT-ANGLE, 2-CYCLE I.C. ENGINE-COMPRESSOR

BY COMBINING vertical gas or diesel engine cylinders with horizontal compressor cylinders, at a right angle, this unit reduces floor space, foundation and shipping costs. Trunk-piston engine driving crankshaft also used for compressor (See cross-section in Prime Mover Section, this issue). Engine is 2-cycle operating on Clark principle of injecting fuel under pressure after cylinder has been scavenged with pure air and exhaust ports have closed. This gives gas consumption no greater than that of common 4-cycle engines. Scavenging air provided by compressor piston incorporated in crosshead of main compressor shaft. Also built with cylinder 1 in. smaller in diameter for conversion to diesel. 2, 3 and 4 power cylinders, with corresponding number of compressor cylinders, and rated 200, 300 and 400 hp. Compressor cylinders for discharge pressure from 50 to 3,000 lb.

*Clark Bros. Co., Olean, N. Y.*





### LOCOMOTIVE-TYPE PORTABLE BOILER

LUCEY, 84-in., Oil Country boiler has no riveted seams in firebox. Arrow shows riveted seam in old-style construction; note absence of rivets in new type. 1,000 and 1,250 sq.ft., 250 and 300 lb. pressures.

*Lucey Boiler & Mfg. Corp., Chattanooga, Tenn.*

**FORDING BY DIESEL**

The generator for the new power plant at North Platte, Neb., weighed 60 tons, so the city wouldn't let it cross the bridges. Instead a Caterpillar diesel tractor forded the river with it

# POWER LINES

## A.S.M.E. Semi-Annual Meeting Program Announced

The A.S.M.E. Semi-Annual Meeting at Dallas, Texas, June 15 to 20, will include following papers and discussion:

*Cracked Residue Fuel Oils* by M. J. Hanlon.
*Analyzing Variable Load for Diesel and Steam Stations* by G. C. Boyer.
Discussion of Relative Merit of Heat Balance Vs. Water-Weighed Boiler Test.
*Why of Boulder Dam* by William Munroe White.
*Heat Transfer and Pressure Drop in Shell and Tube Liquid to Liquid Heat Exchangers* by B. E. Short.
Discussions on A.S.M.E.-A.P.I. Unifired Pressure Vessel Code and Corrosion of Pipe Lines.
*Power and Heat Problems in the Sulphur Mining Industry* by C. L. Orr.
*Chlorination of Circulating Water and Algae Control* by Robert R. Crowdus.
*Operating Experiences with Automatic Combustion Control when Burning Gas and Acid Sludges* by Thomas E. Crossan and A. J. Matherne, Jr.
*Economic Operating Cost Analysis of Several Totally Electrified Pipe Lines* by Winfred H. Stueve.
*Sensible and Latent Heat Control for Air Conditioning* by F. W. Rabe, Jr.
*Compressibility Determination Without Volume Measurements* by E. S. Burnett.
*Design of Diesel Electric Oil Well Drilling Units* by D. M. McCargan and O. A. Hass.
Discussions on High Speed Gas and Diesel Engines for Pumping and Drilling.
*Operating Experience with Pulverized Texas Lignite in a Large Central Station* by Norman G. Hardy.
*Design and Operating Features of the 1450-lb. Deep-Water Steam and Electric Station* by H. G. Hiebeler.
*Utilization of Natural Gas Fuel for Central Stations* by Elmer F. Schmidt.
*Irrigation Pump Equipment and Practice* by B. S. Nelson.

## New Guffey Act Sent to Congress

The Guffey Act for bituminous coal conservation, declared unconstitutional recently by the Supreme Court, will be "partially reborn" when Congress passes a new regulatory measure, apparently approved by the President, similar to the original act but minus the labor provisions. The bill, introduced 48 hours after the Supreme Court decision, retains the interstate price-fixing provisions. The Guffey Act was passed last fall to revive the coal industry after the collapse of the NRA.

## A.I.E.E. Convention In California June 22-26

A.I.E.E. summer convention at Pasadena, Calif., June 22-26, will include the annual business meeting and nine technical sessions in the mornings. During the afternoons, the conference of officers, delegates, and members, two student technical sessions and technical conferences will be held.

Some of the papers to be presented are:

*Present Status of Hydrogen Cooling for Rotating Electrical Machinery* by C. M. Laffoon, Westinghouse Electric & Mfg. Co.
*Hydrogen Cooling — With Near-Critical Velocities* by G. W. Penney, Westinghouse Electric & Mfg. Co.
*Salient Pole Synchronous Motors Operating Out of Synchronism* by A. H. Lauder, General Electric Co.
*Synchronous Mechanical Rectifier-Inverter*—II by S. S. Seyfert, N. S. Hibshman and D. C. Bomberger, Lehigh University.
*Stray Load Tests on Induction Machines*—II by T. H. Morgan and Victor Siegfried, Worcester Polytechnic Institute.
*Measuring Equipment for Oil Power Factor* by L. J. Berberich, Socony-Vacuum Oil Co., Inc.
*The Sparkless Sphere Gap Voltmeter*—II by R. W. Sorensen and Simon Ramo, California Institute of Technology.
*Some Applications of Instrument Transformers* by O. A. Knopp, Pacific Gas & Electric Co.
*Power Transformers for 287.5-Kv. Service* by W. G. James and F. J. Vogel, Westinghouse Electric & Mfg. Co.
*Modern High-Voltage Power Transformers* by K. K. Paluev, General Electric Co.
*Overrefined Oils in Power Transformers* by J. G. Ford, Westinghouse Electric & Mfg. Co.
*The Qualities of Incandescent Lamps* by P. S. Millar, Electrical Testing Laboratories.

TWELFTH NATIONAL EXPOSITION OF POWER AND MECHANICAL ENGINEERING will be held in Grand Central Palace, New York, N. Y., Nov. 30 to Dec. 5, coincident with annual meeting of A.S.M.E.

## NEWS NOTES

LIGHT AND POWER systems in U. S. generated 86,965,000,000 kw.hr. in 1935, and exhibited a total output, including surplus energy, of 102,895,000,000 kw.hr. compared with 73,818,000,000 and 89,323,000,000 in 1933, according to records of 174 hundred-million-kw.hr. public utilities tabulated in a supplement to "Electrical World" of May 9. Numerically, the plants are about equally divided between fuel-burning and water-power, but the kw. rating of the latter is only 29% of the total.

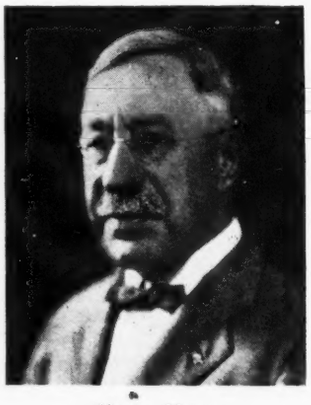
STOKER MANUFACTURERS ASSN. will plan a stoker research program in cooperation with Bituminous Coal Research, Inc., at the annual meeting, June 11 to 13, at White Sulphur Springs, West Virginia.

PRACTICALLY every phase of smoke abatement was covered in papers presented at the 30th annual convention of the Smoke Prevention Association, June 2 to 5, in Atlanta, Ga. T. E. Shaughnessy, Northern Ill. Coal Corp., spoke on results of sulphur in coals; Wm. Calbert, Nashville, Tenn., on smoke abatement in small cities; F. H. Lamping, Cincinnati, Ohio. "Dust Fall and Soot Collection Surveys" and "Practical Smoke Abatement" by J. E. Bjorkholm, Milwaukee, Wis.

32ND annual convention of Association of Iron & Steel Electrical Engineers and Iron & Steel Exposition will be held in Detroit, Sept. 22-25.

A.S.T.M. annual meeting in Atlantic City, June 29 to July 3, will include an address by Dr. H. C. Parmelee, editor, "Engineer and Mining Journal" and the eleventh Edgar Marbury Lecture by Arthur L. Day, director, Geophysical Laboratory, Carnegie Institution of Washington.

EXECUTIVES of eight New Jersey industries and four trunk line railroads reported yearly savings, often greater than the installation cost, brought about by boiler plant improvements, at the Industrial Fuels Conference in Jersey City, recently. William G. Christy, Hudson County Smoke Abatement engineer, presided over the meeting sponsored by the Department of Smoke Regulation.



**Henry Vogt**

REFRIGERATING MACHINERY ASSN. elected Ezra Frick, president of Frick Co., Inc., and Henry Vogt, president of Henry Vogt Machine Co., honorary members at the spring meeting, May 14 to 16, at Hot Springs, Va.

GREAT LAKES EXPOSITION, to be held in Cleveland, Ohio, starting June 27, will mark the hundredth anniversary of Cleveland's incorporation as a city. The exposition will portray industrial, cultural and commercial aspects of the Great Lakes states, emphasizing the story of iron and steel.

ECONOMICS of engineering will be the subject of a summer session of the Society for Promotion of Engineering Education at the Stevens Institute Engineering Camp, Johnsonburg, N. J., during the week beginning June 28. It is proposed to have two morning sessions and one evening session each day from June 28 until Sunday, July 5. Those attending the session will be for the most part teachers in economics and engineering schools and colleges.

AIR CONDITIONING MANUFACTURERS ASSN., Washington, D. C., and E. I. du Pont de Nemours & Co., Wilmington, Del., have become members of Exhibitors Advisory Council, Inc., 230 West 42nd St., New York, N. Y.

PRESENTATION of an air-conditioning application code featured the Air-Conditioning Manufacturers Assn. annual meeting, May 16, in Hot Springs, Va. Officers elected were: pres., J. F. G. Miller, B. F. Sturtevant Co.,; vice-pres., J. A. Harlan, Kelvinator Corp.; treas., P. A. McKitrick, Parks-Cramer Co.

NATIONAL POWER CONFERENCE is the new name of Midwest Power Engineering Conference, 308 W. Washington St., Chicago, Ill. G. E. Pfisterer remains secretary.

## FOREIGN STRAWS

ONTARIO HYDRO-ELECTRIC POWER COMMISSION, Toronto, Ont., is planning to resume development of Trent River hydro projects as a result of cancellation of its contract with the Quebec power companies for surplus power.

LONDON FILM PRODUCTION, LTD., Denham, England, has installed six 1,200-hp., 250-r.p.m., 8-cyl., opposed-piston, 4-cycle, mechanical injection, Crossley-Premier, diesels, driving d.c. generators. 3,000,000 B.t.u. per hr. are recorded by passing exhaust gases through thimble-tube boilers.

MANITOBA POWER COMMISSION, Winnipeg, Man., has approved $200,000 hydro extension for 1936.

CANADIAN WHITE PINE CO., LTD., Kinross St., Vancouver, B. C., is constructing foundations for $15,000 steam power plant.

B. C. PULP & PAPER CO., Vancouver, B. C., is considering additions to power plant at Port Alice mills, Vancouver Island and Woodfibre on Howe Sound.

CITY OF CHARLOTTETOWN, Prince Edward Is., Canada, will construct municipal power plant and distribution system.

CARIBOO GOLD QUARTZ CO., Wells, B. C., will spend approximately $12,000 for a fireproof power-house to replace a frame building recently destroyed by fire.

CANADIAN ELECTRICAL ASSN., Room 408, Power Bldg., Montreal, Que., will hold its 46th Annual Convention at Manor Richelieu, Murray Bay, Que., June 24 to 26.

## Utility Orders

Commonwealth Edison Co. is installing a 1,275-lb., 910-deg. top on its Fisk St. Station, Chicago, Ill. Two 412,500-lb. per hr. boilers have been ordered from Babcock & Wilcox. Order includes superheaters, economizers, air heaters and soot blowers. Sargent & Lundy, consulting engineers.

West Penn Power Co. is also installing a top at Springdale to operate at 1,450 lb. and 935 deg. Three 500,000-lb. per hr. B. & W. boilers will be direct-fired with pulverized coal. Sanderson & Porter, consulting engineers.

Boilers for both plants are special cross-drum, sectional-header type with superheater and economizer sections arranged for divided flow of furnace gas for control of superheat over a wide operating range. Slag-tap, water-cooled furnaces have studded-tube slag screen.

## Diesel Pumping Plant Cuts Cost 75%

North Easton, Mass., automatic diesel pumping station for municipal water service, started last December, is operating at four mills per kw.-hr. The installation consists of a 100-hp., 6-cyl., 4-cycle, Hercules diesel, direct-connected to a single-stage Aldrich centrifugal pump delivering 800 to 1,000 g.p.m. against 170-ft. head at 140 r.p.m.

## OBITUARIES

LEONARD BISHTON-BOTFIELD, 55, president of Botfield Refractories Co., Philadelphia, president of Canadian Botfield Refractories Co., Ltd., Canada., died May 5. Mr. Botfield succeeded his father, A. Bishton-Botfield, as president of the Botfield Refractories Co., in 1916. He was national chairman of the High Temperature Bonding Mortar Division of the American Refractories Institute, and a director of the Institute.

RALPH W. BURNHAM, 50, chief engineer for Eastern Steamship Lines, died April 27.

L. A. BOWDEN, Ohio representative for Jas. P. Marsh Corp., died April 10.

BURTON L. DELACK, general assistant to Vice-Pres. W. R. Burrows of the General Electric Co., and until two years ago manager of the company's Schenectady works, died May 7.

JOHN TEN EYCK HILLHOUSE, 56, retired engineer for Western Electric Co., died recently in Asheville, N. C.

WILLIAM M. MASK, JR., 58, secretary-treasurer of Benjamin F. Shaw Co., died May 12.

HENRY A. MORSS, 65, president and treasurer of Simplex Wire & Cable Co., Cambridge, Mass., died May 6. Mr. Morss was former director and vice-president of Hub Wire Cloth & Wire Works Co., Boston, and director of Arthur D. Little, Inc.

GEORGE D. POGUE, 61, diesel sales engineer and inventor of the helical-spring-hydraulic isolation for the Singer Bldg. (N. Y.) diesel, died May 3. At various times Mr. Pogue had been associated with the American Diesel Engine Co., Fulton Iron Works Co., Busch-Sulzer Bros. Diesel Engine Co., and Winton Engine Corp. During his career he was granted more than 125 patents on mechanical and electrical devices.

S. D'ARCY RICKARD, 49, president of Rickard Engineering Corp., New York, N. Y., died April 9. Mr. Rickard was formerly associated with the Denver Fire Clay Co., and later with Gilbert & Barker Co. at Springfield, Mass. He was particularly interested in oil-burning and storage equipment.

J. H. WHITE, 45, president of Southern National Gas Co., died April 22.

## PERSONALS

GEORGE L. ERWIN, JR., has been appointed assistant to the vice-president in charge of sales for Crane Co., Chicago, Ill.

JAMES M. DEGNAN, formerly with Worthington Pump & Machinery Corp., has been made general sales manager of Hill-Diesel Engine Co., Lansing, Mich.

ROBERT H. BRINTON, P. O. Box 1970, Houston, Texas, has been appointed sales representative for Newark Wire Cloth Co., Newark, N. J.

JOHN A. DAVIS, JR., formerly manager of Ashland district for Virginia Electric & Power Co., Richmond, Va., has succeeded R. G. CARROLL as assistant to the president. Mr. Carroll was promoted to manager of transportation, Norfolk division, succeeding WM. B. MCGORUM, who is now with the Charleston (W. Va.) Transit Co. H. HOLMES HARPER, former member of lighting bureau, has been appointed manager of the Ashland district, succeeding Mr. Davis. J. H. STRIBLING has been made superintendent of the electric department for Richmond district. R. S. WALKER became assistant superintendent electric department for Richmond, and JOS. T. SULLIVAN has been appointed manager for Portsmouth district.

GEORGE CLARK, formerly with Edge Moor Iron Co., has joined the sales staff of Foster Wheeler Corp. at 80 Federal St., Boston, Mass.

ALBERT B. HOFFMAN, formerly chairman of the Board of Detroit Sulphite Pulp & Paper Co., and vice-president of Detroit Trust Co., has been elected president and general manager of Blackmar Rotary Pump Co., Grand Rapids, Mich.

PHILIP C. NEON, vice-president of Borne-Scrymser Co., was tendered a testimonial dinner May 14 upon the completion of 30 years of service with the company.

ALFRED KAUFFMANN has been elected president of Link-Belt Co., 307 N. Michigan Ave., Chicago, Ill., succeeding George P. Torrence, whose resignation as president and director was accepted recently. Mr. Kauffmann has been with the company for 35 years, during which time he has been designing engineer, superintendent of construction, sales engineer, vice-president and manager of the Philadelphia plant, and in charge of the Link-Belt, Dodge and Ewart plants at Indianapolis. For the past four years he has been first vice-president in charge of the Chicago plant. Four board members elected to fill the terms expiring this year are: Wellington Wells, Boston; J. Reece Lewis, Philadelphia; B. A. Gayman, San Francisco; Alfred Kauffmann, Chicago.

E. H. JESSOP, formerly with Morse Chain Co., Ithaca, N. Y., is now in charge of Morse Chain sales for Tranter Mfg. Co., 105 Water St., Pittsburgh, Pa.

E. A. CLEVELAND, newly-elected president of the Engineering Institute of Canada was given a dinner by the Vancouver Branch on his return from the national convention at Hamilton, Ont. Speakers included Major W. G. Swan, W. Powell, Charles Brakenridge, A. S. Wooton and Hugh Mucklestone.

S. C. MERRILL has been promoted to eastern district manager of automotive division for Timken Roller Bearing Co., Canton, Ohio. Headquarters will be in Detroit, Mich.

CHARLES LEE KARR of Scarsdale, N. Y., has been appointed superintendent of power operations for TVA. Mr. Karr was with Detroit Edison Co. for ten years, with Kansas Gas & Electric Co. for seven years and served as operating manager of Carolina Light and Power Co. for four years. His new headquarters will be in the Old Post Office Bldg., Chattanooga, Tenn.

KENNETH MORRISON has been elected vice-president in charge of sales for Mason-Neilan Regulator Co., Boston, Mass.

R. M. GUIRY has been appointed manager of the Chicago office of Kennedy Valve Mfg. Co.

R. J. MURPHY, district representative of Elgin Softener Corp., at Pittsburgh, Pa., has been promoted to division manager with headquarters in Hippodrome Bldg., Cleveland, Ohio. PAUL F. SPARROW has been appointed district representative at Pittsburgh.

WILLIAM H. SANFORD, former chief engineer for Pennsylvania R. R., has been elected secretary-treasurer of the National District Heating Assn., 1317 Spruce St., Philadelphia, Pa.

M. A. CARPENTER has been appointed sales manager for Falk Corp., Milwaukee, Wis., to succeed L. A. Graham, resigned.

GEORGE DANDROW has been appointed manager of the New York district power products and industrial department for Johns-Manville Corp., New York, N. Y.

JAMES E. DE LONG, former vice-president of Waukesha Motor Co., who succeeded H. L. Horning as general manager, has been elected president. James B. Fisher, chief engineer, was elected vice-president.

FLOYD L. GREENE has been advanced to executive vice-president of General Refractories Co., Philadelphia, Pa.

J. S. MCKEIGHAN has been transferred to Dayton-Ohio sales staff of Lincoln Electric Co. J. B. McCormick has been transferred from Philadelphia to the Pacific Coast, and Paul W. James from the factory to Binghamton, N. Y.

H. A. LARSEN, former manager of structural products division on West Coast for Wickwire-Spencer Steel Co., New York, N. Y., has been appointed Pacific Coast sales manager for all products.

L. L. CASKEY has been appointed district sales manager for Republic Steel Corp. in Philadelphia. Mr. Caskey succeeds J. B. DeWolfe, who has been transferred to Cleveland as assistant manager of sales, Tin Plate Division. J. W. BRAFFETT, former Detroit representative of Oliver Iron & Steel Co., has joined the Detroit sales staff.

DR. E. F. LOWRY has resigned as research engineer for Westinghouse Electric & Mfg. Co., to become director of research development for Continental Electric Co.

J. W. LEWIS, former assistant to president Gerard Swope, was elected treasurer of General Electric Co. April 24. He succeeds R. S. Murray, who retired May 1, after 43 years of service.

A. R. MCMULLIN has been appointed manager of the New York office, Heavy Machinery Division, Nordberg Mfg. Co., Milwaukee, Wis.

MUIR L. FREY, former chief metallurgist for John Deere Tractor Co., Waterloo, Ia., has joined the metallurgical staff of Republic Steel Corp., Buffalo, N. Y.

J. O. DONOHUE, Baltimore, Md., and J. D. MCLIN, Dallas, Tex., have rejoined the sales staff of New York Belting & Packing Co., Passaic, N. J.

A. D. HEFFRON, JR., has been appointed district sales manager for Babcock & Wilcox Tube Co. at 1502 Marquette Bldg., Chicago, Ill.

W. R. RAMSAUR has been appointed chief engineer of Young Radiator Co., Racine, Wis. Warren Ewald has been added to the staff of this company.

MORRIS E. LEEDS, president Leeds & Northrup Co., Philadelphia, Pa., was awarded the Henry Laurence Gantt Gold Medal "for distinguished achievement in industrial management as a service to the community," at the annual meeting of the American Management Assn., June 4 and 5, in New York.

M. ROGERS has been appointed general factory manager of Caterpillar Tractor Co., Peoria, Ill. JAMES R. MUNRO succeeds Mr. Rogers as factory manager of the tractor division.

E. HESTON OWENS has been appointed sales engineer in the New York Office of Paul B. Huyette Co., Philadelphia, Pa.

S. C. PARTRIDGE has been appointed assistant general manager of industrial division, with headquarters in Canton, for Timken Roller Bearing Co., Canton, Ohio.

PROF. LIONEL S. MARKS, mechanical engineering dept., Harvard Engrg. School, who was injured in an automobile accident May 8, is expected to recover.

GEORGE E. OLMSTEAD, New Haven, Conn., has joined sales staff of Morse Chain Co., Ithaca, N. Y.

E. T. BENNINGTON, formerly with Cleveland Tramrail Co., has become sales engineer for Harnischfeger Corp., in Chicago.

L. G. ATKINS, formerly superintendent of steam and power department of Saltville operations of Mathieson Alkali Works, Inc., has become affiliated with Southern Chemical Corp., Charlotte, N. C. J. C. PORTER succeeds Mr. Atkins.

L. E. MUSTARD has been appointed district manager of Bristol Co.'s branch office in Detroit.

DR. HERBERT VICKERS, head of the department of mechanical engineering will leave the University of British Columbia, Vancouver, B. C., at the end of the present term to accept a post with an English electrical engineering company.

GEORGE P. SCHUMACKER, formerly iwth Worthington Pump & Machinery Corp. has opened an office at 1120 Chester Ave., Cleveland, Ohio, representing Cooling Tower Co., Inc., Pennsylvania Pump & Compressor Co., Quincy Compressor Co., National Steam Pump Co., Stets Co., V. D. Anderson Co., Williams Valve Co., Sea-Ro Packing Co. and Mabbs Hydraulic Packing Co.

EVANS L. SHUFF, 686 Greenwood Ave., Atlanta, Ga., has been appointed sales representative for Moore Steam Turbine Corp., Wellsville, N. Y.

PHILIP A. SINGLETON, Philadelphia, Pa., and JACK K. WILLIAMS, Chicago, Ill., have been added to the sales force of Reliance Electric & Engrg. Co., Cleveland.

R. J. WAGNER, formerly connected with D. O. James Mfg. Co., Chicago, Ill., has been added to Chicago sales force of Machinery & Welder Corp., and H. H. PORTER, formerly with O. H. Dittmer Co., Milwaukee, Wis., has been added to the Milwaukee sales force.

W. R. WOOD, former sales engineer for Combustion Engrg. Co., is now connected with Plibrico Jointless Fire Brick Co., 1800 Block, Kingsbury St., Chicago, Ill.

WILLIAM S. SHIPLEY, president of York Ice Machinery Corp., and George W. Mason, president of Kelvinator Corp., will represent the A.S.R.E. at the World Power Conference.

C. H. TATE has been appointed Cleveland district representative for Northern Equipment Co., Erie, Pa., at 507 Hunkin-Conkey Bldg., 1740 East Twelfth St., Cleveland.

WALTER LEEK of Leek & Co., Ltd., heating engineers, Vancouver, B. C., was elected president of the Vancouver Building & Construction Industries Exchange by acclamation at the annual meeting of the organization, held recently. F. J. Hale, Empire Sheet & Metal Works, Ltd., was elected first vice-president. Other officers are Col. J. F. Keen, E. J. Ryan Contracting Co., second vice-president; E. C. Dougherty, Continental Marble Co., Ltd.; A. S. Gentles, general manager Dominion Bridge Co., Ltd., and J. A. Baxter, The O'Neil Co., Ltd., directors for two years, and Col. S. J. Anderson of Central Sheet Metal Works, Ltd., for one year.

T. H. VADEN, Montgomery, Ala., has been appointed southern district superintendent for Alabama Power Co. Mr. Vaden has been assistant division superintendent for some time.

EDWIN H. SCHMITZ, formerly with Riley Stoker Corp., has joined C. H. Sprague & Co., Boston, Mass., as combustion engineer.

F. B. YATES has been appointed manager of New York district office in charge of industrial sales for Timken Roller Bearing Co., Canton, Ohio.

LEE W. TREMBLAY, Witherspoon Bldg., Philadelphia, Pa., has been appointed sales representative for Edward Valve & Mfg. Co., Inc., East Chicago, Ind.

A. D. HEATH, 1701 N. Illinois St., Indianapolis, Ind., has been appointed representative for Ex-Cell-O Aircraft & Tool Co., Detroit, Mich., in southern Indiana.

EDWARD P. PREZZANO was elected president of Westchester Lighting Co., Yonkers Light & Power Co. and Bronx Gas & Electric Co. to succeed the late Eugene Rosenquest. Floyd Carlisle was elected chairman of the board of the Westchester company. Mr. Prezzano has been vice president of the company for the past ten years and a member of the board of directors. He was also vice president and secretary of the Yonkers and Bronx companies. He was recently made president of the Tarrytown Terminal Corp., which is also a part of the Consolidated system.

## BUSINESS NOTES

SULZER BROS., Winterthur, Switzerland, has licensed American Locomotive Co., 30 Church St., New York, N. Y., to manufacture, use and sell Sulzer 2-cycle diesel engines. Engines will be built in America by McIntosh & Seymour Corp., Auburn, N. Y., wholly-owned subsidiary of American Locomotive Co. under the trade name, "Alco-Sulzer."

INTERNATIONAL NICKEL CO., INC. has opened an office at 333 N. Michigan Ave. Bldg., Chicago, Ill., under H. L. Geiger and one at 705 Petroleum Securities Bldg., Los Angeles, Calif., under A. G. Zima.

CUTLER-HAMMER, INC., Milwaukee, Wis., has opened an office at 10 West Chase St., Baltimore, Md., under R. A. Haworth.

TRUNDLE ENGRG. CO., Cleveland, Ohio, will assist Cummins Engine Co., diesel engine manufacturers of Columbia, Ind., to work out an expansion program. .

REPUBLIC STEEL CO., Cleveland, Ohio, has appointed EQUITABLE EQUIPMENT CO., INC., New Orleans, La., distributor for stainless steel; YORK CORRUGATING CO., York, Pa., and SHEET METAL MFG. CO., INC., Stamford, Conn., iron sheet distributors.

NORTH AMERICAN REFRACTORIES CO., Cleveland, Ohio, has acquired the silica brick plant and property of Lavino Refractories Co. formerly owned by E. K. Lavino and Co. at Womelsdorf, Pa. No other property of E. J. Lavino & Co. is included.

F. R. MERRILL & CO., 250 Park Ave., New York, N. Y., has been appointed representative for Moore Steam Turbine Corp., Wellsville, N. Y.

LINCOLN ELECTRIC CO., Cleveland, Ohio, has moved its Pittsburgh office from 323 Fourth Ave., to 926 Manchester Blvd.

FOSTER WHEELER CORP. and GENERAL REGULATOR CORP. have consolidated their sales departments under C. J. King, 165 Broadway, New York, N. Y.

ROLAND H. BAKER, contractor, has opened an office at 10 State St., Boston, Mass.

JAMES T. GORDON CO. has moved to 11 Park Place, New York, N. Y.

HENDRICK MFG. CO.'s branch office, 30 Church St., will handle the New York sales of "Mitco" products.

CHAIN BELT CO. has moved its Pittsburgh office to Room 526, Grant Bldg. G. E. Mahoney, former sales engineer in the Milwaukee office, will be in the Pittsburgh office.

**40,000-kw. ELEPHANT**

Outer high-pressure shell casting of a 40,000-kw. steam turbine presents this elephant-like appearance as it emerges from the mold before cleaning, trimming and machining at the G.E. Schenectady works.

ALLEN-BRADLEY CO. has moved its New York office from 50 Church St. to 30 Vesey St.

STANDARD FUEL ENGINEERING CO., 667 Post Ave., S., Detroit, Mich., has started construction of a new addition to its plant.

E. CORY & CO., Portland, Me., have been appointed agents for General Refractories Co., Philadelphia, Pa.

LINK-BELT CO., New York, N. Y., has appointed Tulio Ospina y Cia., Medellin, Colombia, representatives in the Republic of Colombia.

AMERICAN ENGINEERING CO. has moved its New York sales office from 40 West 40th St., to Evening Post Bldg., 75 West St., New York, N. Y.

DEWOLF FURNACE CORP., Rochester, N. Y., has opened an office at 90 West St., New York, N. Y., under William F. Wurster.

ASKANIA REGULATOR CO., 1603 S. Michigan Ave., Chicago, Ill., has taken over automatic regulator business of H. A. Brassert & Co., and American Askania Corp.

GENERAL REFRACTORIES CO., 106 S. 16th St., Philadelphia, Pa., has appointed BRAMAN, DOW & CO., Boston, Mass., dealer agents for Boston.

PENNSYLVANIA PUMP & COMPRESSOR CO., Easton, Pa., has appointed Lieb-Jackson, Inc., 337 South High St., Columbus, Ohio, and F. B. Schwartz, 1019 Washington Ave., So., Minneapolis, Minn., agents in their respective territories.

WESTINGHOUSE ELECTRIC & MFG. CO. has moved its New York executive and sales office to the Westinghouse Bldg., 150 Broadway, New York, N. Y.

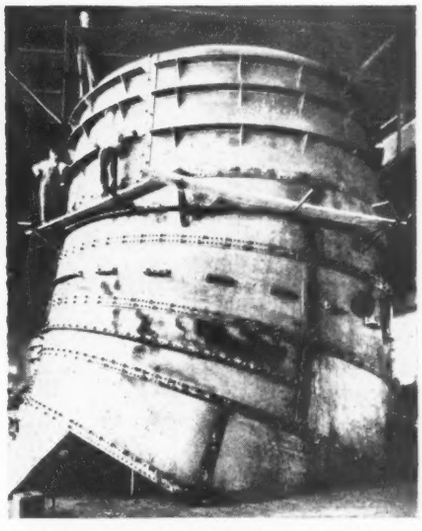
ROOTS-CONNERSVILLE BLOWER CORP., Connersville, Ind. has moved its Chicago office under William Townsend, district manager, to Marquette Bldg., 140 S. Dearborn St.

FOXBORO CO. has moved its New York office to the Graybar Bldg., 420 Lexington Ave., New York, N. Y.

GATES RUBBER CO., Denver, Colo., has opened a regional warehouse and sales office at 71-73 Murray St., New York, N. Y., called The Gates Co.

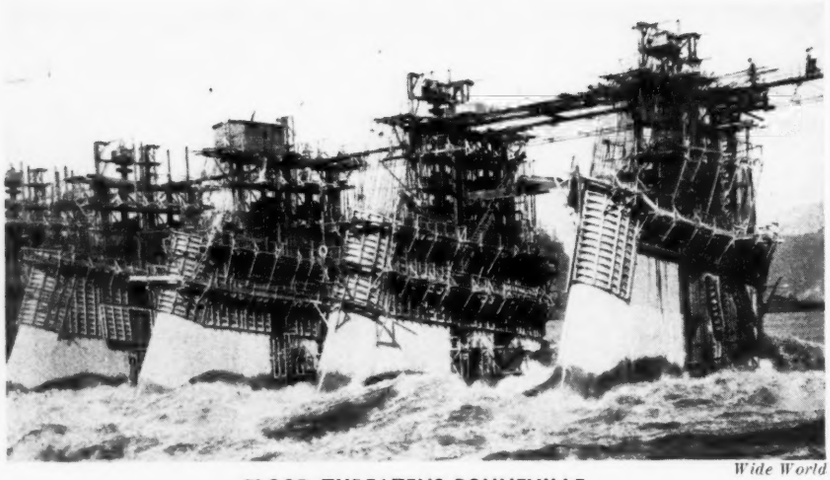
REPUBLIC STEEL CORP. has opened a new sub office in the State Bank Bldg., Albany, N. Y., with J. M. Higinbotham, salesman in charge.

AMERICAN UNIFORM BOILER-LAW SOCIETY has moved to Singer Bldg. Annex, 95 Liberty St., New York, N. Y.



**LINER FOR LARGEST KAPLAN WHEEL**

Draft tube liner for one of two 66,000-hp. Kaplan wheels being built by S. Morgan Smith Co., for the Bonneville development on the Columbia River. The draft tube liner is 23 ft. 4 in. in diameter at the top, which is the approximate diameter of the runner, is 25 ft. high, weighs 130,000 lb., and is made of ¾-in. steel plate. These units are the highest-powered Kaplan wheels in the world.



*Wide World*

**FLOOD THREATENS BONNEVILLE**

Inspired efforts of workmen saved all but a small part of the cement forms from being carried away when 10-ft. waves threatened the partially completed Bonneville Dam on the Columbia River east of Portland, Ore. Two 66,000-hp. Kaplan turbines are being installed to operate under a head of 67 to 22 ft. in the plant which will have an ultimate capacity of 665,000 hp.

# STRAWS

## Pointing the way business winds blow

GRAND RAPIDS (Mich.) GAS LIGHT CO., has authorized 10¾-in. welded-steel pipeline from Mecosta-Montcalm Counties gas field to Grand Rapids, about 46 miles. Will install mixing plant for natural and artificial gas in city. Cost over $300,000.

KENTUCKY NATURAL GAS CORP., Owensboro, Ky., has authorized 12-in. welded-steel pipeline from Terre Haute, Ind., to Montezuma, Ind., about 25 miles. Line will connect with system of Panhandle Eastern Pipe Line Co., Kansas City, Mo. Installation will be by Connor & Son Construction Co., City Bank Bldg., Kansas City. Cost over $175,000.

GUNNISON, COLO., contracted Hendrie & Bolthoff Mfg. Co., 1639 17th St., Denver, Colo., for engine-generator, boilers and auxiliary equipment for municipal electric plant. F. E. Heenan, city manager, in charge.

U. S. ENGINEER OFFICE, Portland, Ore., has low bid from General Electric Co., Schenectady, N. Y., for 2,400-volt metalclad switchgear for New Bonneville, Ore., hydro plant, at $77,949.

FARIBAULT (Minn.) WOOLEN MILLS plan steam power house for textile mill to include 200-hp. high-pressure boiler and auxiliary equipment. Diesel station is under consideration. Ralph W. Richardson, New York Bldg., St. Paul, Minn., consulting engineer.

GREENFIELD, OHIO, has revised plans for extensions in municipal power plant and waterworks to include 1000-kw. turbo-generator and accessories. Board of Public Affairs in charge. W. I. Barrows, Reibold Bldg., Dayton, Ohio, consulting engineer.

EAST LIVERPOOL, OHIO, plans municipal electric plant. $1,500,000 secured through PWA. Peter F. Loftus, Oliver Bldg., Pittsburgh, Pa., consulting engineer.

BUREAU OF RECLAMATION, Austin, Tex., has low bid for construction of hydroelectric station, Colorado River Project, Tex., from Morrison-Knudsen Co., Boise, Idaho, at $783,773.

NEWPORT (Vt.) ELECTRIC CORP. has contracted Vappi & McDonald Co., 240 Sidney St., Cambridge, Mass., for hydroelectric station, including penstock and auxiliary structures Cost over $100,000. Charles T. Main, Inc., 201 Devenshire St., Boston, Mass., consulting engineer.

VERMILLION, S. D., has contracted Worthington Pump & Machinery Corp., Harrison, N. J., for equipment for municipal power plant at $52,247. J. W. Kolb., city engineer, in charge.

CABOT GAS CORP., Olean, N. Y., subsidiary of Godfrey L. Cabot, Inc., 77 Franklin St., Boston, Mass., has contracted Eastman Kodak Co., Kodak Park, Rochester, N. Y., for natural-gas supply. Plans construction of 14-in. welded-steel pipeline from Western Pennsylvania field, about 100 miles. Will cost approximately $1,000,000. Eastman will use natural gas instead of coal for boilers.

SOUTHERN PACIFIC RAILROAD, Portland, Ore., plans cold storage plant at Brooks, Ore., 60 x 170 ft., to cost over $45,000. Ertz & Burns, Pittock Block, Portland, architects.

PORTSMOUTH, VA., has authorized surveys and estimates for municipal electric plant. E. B. Hawks, city manager.

STATE BOARD OF EDUCATION, Indianapolis, Ind., plans power plant at school near Fort Wayne, Ind. Bids will be asked in July. Cost about $100,000 with boilers, pumping machinery and auxiliary equipment. Pohlmeyer & Pohlmeyer, Central Bldg., Fort Wayne, architects.

SINCLAIR-PRAIRIE CO., Independence, Kan., plans 8-in. welded-steel pipeline from Ada, Okla., to Pauls Valley, Okla., about 42 miles, for oil. Cost $250,000.

DAUPHIN, MAN., Canada, will vote on $75,000 addition to power plant. R. Fagen, municipal clerk.

PUBLIC SERVICE CO. OF COLORADO will extend its Valmont plant to install 34,000-hp. reserve capacity at cost of about $1,200,000.

UNITED GAS SYSTEM, Rusk Bldg., Houston, Tex., plans extentions in gas booster plant in Rodessa, La., oil and gas field. Will also add to gasoline absorption plant at Myrtis with installation of compressors, tanks and other equipment. Will cost over $500,000.

STATE DEPT. OF INSTITUTIONS, Springfield, Ohio, has low bid from Elliott Co., Chicago, Ill., for engine and generator units for State Hospital, Kankakee, Ill.

TVA WHEELER DAM power plant will be equipped with School Choffin Co., Youngstown, Ohio, ventilating systems to cost about $6,600.

TIMKEN ROLLER BEARING CO. has ordered four Taylor stokers, two for 3,070-sq.ft. and two for 3,940-sq.ft. boilers.

DETROIT EDISON CO. has ordered its seventh double-opposed Taylor stoker for a 330,000-lb. per hr. boiler.

NATIONAL SUPERIOR CO., Springfield, Ohio, subsidiary of National Supply Co., Toledo, Ohio, has begun work on a plant addition costing more than $100,000.

YOUNGSTOWN SHEET & TUBE CO. Youngstown, Ohio, will install high back-pressure turbine at Campbell mills.

LA CROSSE, KANSAS, municipal electric plant which operates three De La Vergne diesels totaling 710 hp., reports an average operating cost of 1.1 cents per kw.-hr. for the past 15 years. Since 1921 the plant has generated 10,184,906 kw.-hr., developing 9.66 kw.-hr. per gal. (including fuel for heating office), spending 92 cents per hp. per year for repairs and supplies.

TEXTILE ICE & FUEL CO., Greenville, S. C., will make extensions in ice-manufacturing and cold storage plants, including addition to plant on Buncombe Rd., and removal of plant on West Washington St. Single unit will be operated for capacity of about 30 tons per day. Cost about $45,000. C. F. Cox, president.

MOORHEAD, MINN., has contracted T. F. Powers Construction Co., 217 Fourth Ave., North, Fargo, N. D., for addition to municipal power plant. Cost close to $100,000, with equipment. Ralph D. Thomas, 1200 Second Ave., South, Minneapolis, Minn., consulting engineer.

SOUTH CHARLESTON, OHIO, is considering municipal electric plant to cost over $75,000. C. D. Juvenal, mayor.

HYATT ROLLER BEARING CO., subsidiary of General Motors, will abandon its Harrison, N. J., generating plant and purchase power from Public Service Electric & Gas Co. Connected load of 7,500 hp. is expected to consume more than 1,000,000 kw.hr. per month.

BOILER and engine industry will receive approximately $2,300,000 of the $175,000,000 Naval appropriation for the next ten years. Other industries benefiting are: Steel, $15,500,000; heavy machinery, $25,000,000; pipe tubing, $1,400,000; electrical cable and wire, $800,000; electrical apparatus, $3,000,000; heat insulating materials, $350,000.

LOUP RIVER PUBLIC POWER DISTRICT, 2307 Thirteenth St., Columbus, Neb., Harold Kramer, secretary, plans electric-operated pumping plant, with two main units and auxiliary equipment, in connection with desilting works and intake system for hydroelectric project. Will include traveling bridge, sluice gates and large settling basin, and will cost $1,300,000. $7,300,000 secured through Federal aid. Harza Engrg. Co., 205 W. Wacker Drive, Chicago, Ill., consulting engineer.

BINGHAMTON, N. Y., will purchase pumping machinery and accessory equipment for nine pumping plants for municipal waterworks. Cost close to $100,000. J. A. Giles, City Hall, city engineer.

CHEROKEE VINEYARD ASSN., Acampo (San Joaquin County), Calif., plans steam plant at winery at Georgia Station, near Acampo to cost about $145,000. Peter Sala, 2130 North Commerce St., Stockton, Calif., architect.

J. A. DARBY, 1718 Banks St., Houston, Tex., heads project to construct 75-ton ice-manufacturing plant at Miller and Leeland Sts. Building contract let to M. K. Bercaw, 830 Arlington St. Cost about $85,000.

AUSTIN, MINN., contracted A. R. Robertson Co., 1603 University Ave., St. Paul, Minn., for addition to municipal electric plant, with installation of boiler, stoker, air preheater and auxiliaries. Cost about $130,000. Ralph D. Thomas, 1200 Second Ave., South, Minneapolis, Minn., consulting engineer.

VILLISCA, IOWA, plans new call for bids for municipal electric plant following court ruling. Will include two diesels and accessories. $150,000 available. Burns & McDonnell Engrg. Co., 107 West Linwood Blvd., Kansas City, Mo., consulting engineer.

WASHINGTON, KAN., plans municipal diesel-electric plant. $100,000 arranged through Federal aid. E. T. Archer & Co., New England Bldg., Kansas City, Mo., consulting engineers.

GRAYS HARBOR PULP & PAPER CO., Hoquiam, Wash., plans boiler plant at local mill and will begin work soon. Cost close to $50,000.

DOVER, OHIO, will take bids in June for addition to municipal electric plant, expansion in steam division, including two boilers and accessories. Cost about $70,000. Rosch & Hartline, New Philadelphia, Ohio, consulting engineers. Homer Keppler, director, department of city service.

CEDARBURG, WIS., is considering extensions in municipal electric plant, including 750 or 500-kw. diesel. Light and Water Commission, A. W. Robken, chairman.

STATE BOARD OF ADMINISTRATION, Bismarck, N. D., has let contract to Lignite Combustion Engrg. Corp., 304 Main St., Bismarck, for two boilers for state institution at San Haven, N. D., at $57,950. Coal and ash-handling machinery and other equipment will also be installed. William F. Kurke, 1117 Thirteenth Ave., North, Fargo, N. D., architect.

CUDAHY PACKING CO., Chicago, Ill., plans steam plant at branch meat-packing plant at East Albany, Ga. Cost about $400,000.

CONTINENTAL OIL CO., Ponca City, Okla., has authorized surveys for 4-in. welded-steel pipeline from Valley Center, Kan., to Burrton, Kan., about 16 miles, for crude oil transmission. Two large booster plants will be built at Genesco, Kan., and Moundridge, Kan., for another line. Cost over $150,000. R. T. Looney, Ponca City, engineer.

BOEING AIRCRAFT CO., Seattle, Wash., will build one-story power plant, 35 x 50 ft., at assembling works at East Marginal Way and 14th Ave. Contract has been let to Austin Co., Dexter-Horton Bldg., for about $250,000.

MINNEAPOLIS-MOLINE POWER IMPLEMENT CO., Minnehaha Ave., Minneapolis, Minn., plans steam plant at Hopkins, Minn., to include three boilers and accessories. Cost about $125,000.

MISSION SPRINGS DISTILLING CO., INC., 406 Republic Bldg., Louisville, Ky., plans steam plant at Vine Grove, Ky., which will cost about $175,000.

CONSOLIDATED EDISON CO., New York, N. Y., has authorized extensions and improvements in coke-handling system at East 110th St. plant to cost close to $50,000.

FORD MOTOR CO., Dearborn, Mich., is working on pumping station at River Rouge plant for total output of 140,000 gals. per min. Chlorination equipment will be installed. Cost about $300,000.

MINDEN CO., St. David, Ill., contracted with Patterson & Hartrich, 105 West Adams St., Chicago, Ill., for steam plant at coal-mine. Cost about $45,000. C. M. Garland & Co., 38 S. Dearborn St., Chicago, consulting engineers.

STATE PRISON BOARD, Denver, Colo., plans extensions in hydroelectric station at State Penitentiary, Canon City, Colo.

G. H. KITTREDGE, 8100 E. Jefferson St., Detroit, Mich., heads project to construct and operate refrigerating and precooling plant on waterfront at Fort Lauderdale, Fla. Cost about $100,000, with equipment.

SHELL OIL CO., San Francisco, Calif., soon takes bids for 10-in. welded-steel pipeline from Kern County oil field to refinery at Martinez, Calif., about 260 miles. 4 booster stations will be constructed and existing plants between Caliloa and Martinez will be remodeled and expanded. About 44 miles of steel pipeline will be installed in oil-field gathering system. Will purchase about 25,000 tons of pipe which will cost about $4,500,000. J. U. Stair, vice-president in charge of production; N. W. Thompson, chief engineer.

BECKETT PAPER CO., Hamilton, Ohio, let contract to F. K. Vaughn Bldg. Co., Hamilton, for power house at mill. Also for high-pressure boilers to Babcock & Wilcox Co., New York, N. Y.; and for concrete stack to Rust Engrg. Co., Pittsburgh, Pa. Cost about $200,000. Fosdick & Hilmer, Union Trust Bldg., Cincinnati, Ohio, consulting engineers.

STATE BUILDING COMMISSION, Jefferson City, Mo., has following low bids for equipment at institutions at Fulton, Farmington, Nevada, Marshall and Mount Vernon, Mo.: Elliott Co., and W. O. Weidener, Kansas City, Mo., joint bid, engine-generator units with accessores, $107,301 each, for Fulton, Farmington, Marshall and Mount Vernon; Laclede Stoker Co., St. Louis, stokers, $20,498 each, for Farmington and Mount Vernon; Murray Iron Works, Burlington, Iowa, for boilers, $41,003 each, for Marshall and Mt. Vernon, and for engine-generator unit for Nevada, $15,543,; Consolidated Chimney Co., Chicago, Ill., for stacks, $13,140 each, for Farmington and Fulton plants. Charles A. Haskins, Finance Bldg., Kansas City, Mo., supervising engineer.

LIBBEY-OWENS-FORD GLASS CO., Toledo, Ohio, plans steam power house at sheet-glass plant at East Toledo, to cost over $100,000.

VALLEY PIPE LINE CO., Los Angeles, Calif. (Joint interest of Texas Corp. and Seaboard Oil Corp.), has authorized 10-in. welded-steel pipeline from Kettleman Hills oil field to Estero Bay, Calif., about 70 miles. Pipe line will have capacity of 15,000-bbl. per day and cost $1,500,000. Installation by Lang Transportation Corp., 5501 Santa Fe Ave., Vernon, Los Angeles.

CLIMAX MOLYBDENUM CO., Climax, Colo., plans addition to boiler house at mining and milling plant. Cost close to $75,000 with boilers and accessories. G. Meredith Musick, Republic Bldg., Denver, Colo., architect and engineer. Company headquarters at 500 Fifth Ave., New York, N. Y.

TARBORO, N. C., plans additional equipment at municipal power plant, including diesels and accessories.

PELLA, IOWA, has contracted Lautenbach & Son, Pella, for municipal electric plant addition.

PICAYUNE (Miss.) ICE CO., plans rebuilding portion of ice-manufacturing plant recently destroyed by fire. Cost close to $40,000. Charles F. Rantz, New Orleans, La., consulting engineer.

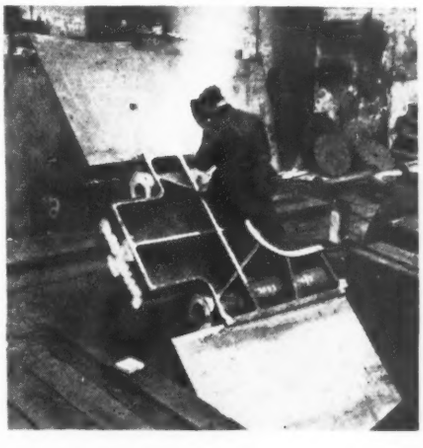
BUFFALO, WYO., is considering municipal power plant. Estimates being made. Will ask bids soon.

LONG PRAIRIE, MINN., has authorized surveys and estimates for municipal electric plant. Burlingame & Hitchcock, Sexton Bldg., Minneapolis, Minn., consulting engineers.

DIFCO LABS., INC., 920 Henry St., Detroit, Mich., plans addition to boiler plant to cost close to $30,000. Donaldson & Meier, Washington Blvd. Bldg., architects.

## PLATE TO CASTING

**Double the power in the same space resulted when design of a 4,000-hp., double-acting, 2-stroke, 4-cyl., 27½x47½-in. diesel was altered to permit use of steel plate and arc welding. American technique of combining steel castings and steel plate by welding was adopted as shown, the bedplate, of which this is a part, weighing only 29¼ tons, as compared with 44 tons for a casting. Lincoln Electric photo**



# SMOKING STACKS . . .

"MEN are working again. Factory wheels are turning, people are buying, chimneys are smoking once more." So declaim newspaper editors about returning prosperity. The bane of cities a few years ago now is welcomed with open arms, for it means that men are going back to work.

But those smoking "chimneys," particularly on power plants, mean nothing more nor less than waste. Down under them somewhere is inefficient combustion, obsolete firing methods that are making somebody's hard-won dollars go up the stack.

The danger of smoke to people is a real one, but one that can never be solved until the average man realizes that most of the smoke and dirt that gets into the air is not coming from power plants, but from poorly and improperly fired boilers in apartments and private houses. The power engineer knows—or should know—better, for he must show efficient use of the heat units in the fuel he burns.

Yet, not so long ago, I heard of an industrial manager who, confronted by a warning from the City Smoke Inspector, instructed his engineer to heat up pickling vats and process tanks before dawn in the morning. *That* was the smoke that he had been warned about, so if it emerged from his stack at night, no one would be the wiser.

But what was really wrong was our old friend obsolescence. Those old boilers were hand-fired, with the grates too close to the boiler shell. Every time the fireman threw in a shovelful of coal, a good part of it went up the stack as smoke, never doing anybody any good. The same industrial manager probably prided himself on his efficient production, his low manufacturing costs. Had he been accused of throwing company dollars into the river, he would have been properly insulted, but when they went up the stack he didn't miss them.

There's an old saying that "where there's smoke there must be some fire." Right! And where there's smoke, there's also inefficiency, dollars wasted in a mistaken idea that old-fashioned methods are cheaper than some of these new-fangled ideas for saving coal. But check up on these "new-fangled ideas"—most of them are time-tried and very practical. Look through the dozens of examples in these pages of money saved be intelligent modernization. It's best to remember that there may be fire without smoke, steam generation without that "cheering black plume" at the top of the stack.

George Edwards

GEORGE EDWARDS,
*Engineer.*